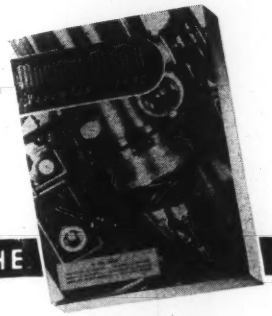

This Month's Cover: Head of Moore jig grinder. Design features of this machine are discussed and illustrated in the article beginning on Page 104.

MACHINE DESIGN

Vol. 22—No. 11 November, 1950

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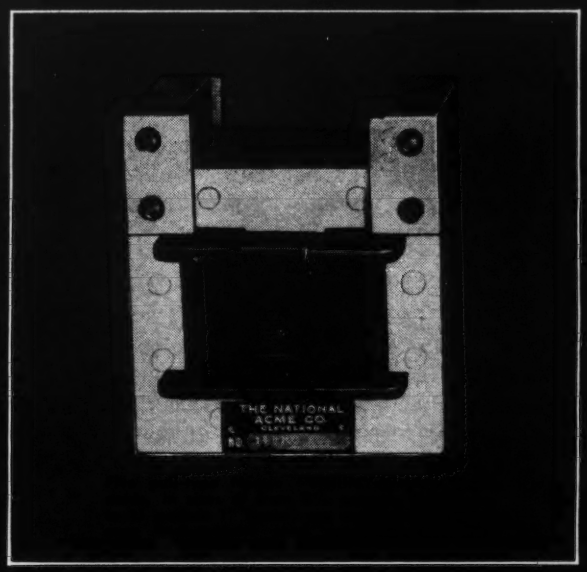
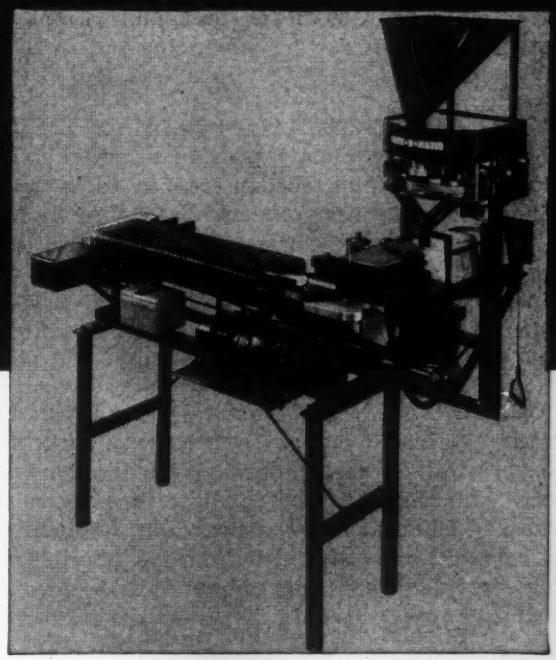


Plageman Picks NAMCO Stellite-Welded SOLENOIDS



The Pack-A-Matic automatic seed packaging machine manufactured by Plageman Enterprises, Willoughby, Ohio, has a capacity of up to 4500 packages per hour.

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TOPICS

LIQUID NITROGEN is being used in a process developed by Linde Air Products Co. for the impact pulverization of hard-to-grind materials such as plastics, pharmaceuticals, insecticides, foodstuffs, etc. In the new technique, liquid nitrogen in spray form cools the material rapidly to a point of maximum fragility just before it enters the pulverizing mill.

THREE-DIMENSIONAL SCALE MODELS cut from a new plastic provide a portrait in color of the strains encountered in tools, machine parts and other objects. Developed by M. M. Leven and H. F. Minter of Westinghouse, the new "photo-plastic" can be cast in sizes up to 8 inches in diameter (or square) and 36 inches in length.

COATING OF ALUMINUM by the Pylumin Process, patented by the Pyrene Manufacturing Co., provides corrosion resistance without additional painting and also serves as an adhesive base for paint finishes. Applied by immersion in heated tanks, the new coating requires a single powder whose ingredients have been proportioned and premixed.

A NEW STAINLESS STEEL has been developed by the Carnegie-Illinois Steel Corp. for metal-glass television tubes. Having thermal properties paralleling those of glass, the new alloy contains 17 per cent chromium.

CAST MAGNESIUM WINGS for aircraft are being investigated by Northrop Aircraft Inc. The outer half of a wing will be cast in one piece with a view to obtaining freedom from porosity and adherence to close tolerances.

TINY GERMANIUM PHOTOCELLS of three types were recently described by George D. O'Neill of Sylvania Electric Products Inc. One type, the photoresistance cell, is able to produce electric signals at least as great as 25 volts when a light shining on the cell is interrupted or modulated. Another, the photoconductance cell, has a current response large enough to operate a relay directly in applications such as alarms, door openers, etc. The third type, the photovoltaic cell, is similar to a tiny battery but has the peculiar property of supplying a voltage in proportion to the light shining on the germanium.

AUTOMATIC PARACHUTE for high-altitude bailouts has been developed by USAF. Taking some of the hazards out of free-fall jumps, the new parachute is controlled by both time and altitude delays which are preset by the pilot before takeoff. The standard rip cord has been retained and may be operated independently of the automatic release.

MAGNETIC CERAMIC having ten times the permeability of previous powdered-iron cores has been introduced by Henry L. Crowley & Co. under the name Croloy. Although applications are still being developed, Croloy is particularly suited to transformers and deflection yokes for TV sets and as a replacement for the usual loop antenna in radio sets. Croloy is also an excellent insulator.

MOLYBDENUM - BASE LUBRICANTS in nonvolatile-liquid form have been announced by the Lockrey Co. Supplementing a line of Liqui-Moly lubricants of volatile-liquid form, the new types remain liquid at all temperatures in applications where a circulating-cooling effect is required. By its inherently "greasy" character and its molecular attraction to bearing surfaces, molybdenum effectively lubricates at extremes of temperature and pressure.

ZIPPERS are being sported by the latest military airplanes as fasteners for the fabric flaps used as seals between surfaces of ailerons and tail sections. According to B. F. Goodrich Co., the rubber-lipped zippers provide a seal against liquids and gases from zero pressure up to the structural strength of the fastener.

COPPER-CLAD STEEL known as Electroshield is being produced by American Cladmetals Co. for the shielding of electronic communication equipment. Consisting of a core of mild steel clad on both sides with copper, the new material can be worked and welded easily.

MACHINE DESIGN
A PENTON PUBLICATION

Design Competence—A Key

FOR AN excellent criterion by which to judge the competence of machine designers we can go back more than three-quarters of a century, when Oliver Wendell Holmes observed that there are "one-story intellects, two-story intellects, three-story intellects with skylights. All fact-collectors, who have no aim beyond their facts, are one-story men. Two-story men compare, reason, generalize, using the labors of the fact-collectors as well as their own. Three-story men idealize, imagine, predict; their best illumination comes from above, through the skylight."

Engineering executives might well paste this observation in their hats. It cuts to the heart of design competence through the maze of qualifications which too often are mistaken for competence itself—such as scientific education, mathematical ability, or practical experience. By themselves these factors do not make a top-story intellect. They are only windows which shed light on problems—like research laboratory apparatus which, as Kettering has pointed out, can only change a mental attitude, never solve a problem. Obviously, the more windows there are the more light is shed, but the "best illumination comes from above, through the skylight."

Viewed in this way, most arguments as to what education, experience or abilities to look for assume their proper perspective. Regardless of formal qualifications, a first or second-story man will be no better than mediocre as a designer. The creative intellect of the third-story man is the essential ingredient of a competent design engineer.

Colin Carmichael

EDITOR

Designing a Modern Machine

Jig grinder incorporates pneumatic, hydraulic, and electric circuits with pushbutton controls

By Frederick C. Victory
Chief Engineer
Moore Special Tool Co.
Bridgeport, Conn.

WHEN first conceived and produced by Moore twelve years ago, the jig grinder was designed primarily to position accurately and grind cylindrical as well as conical holes, with taper in either direction. Since that time, continuing development and user-experience have revealed a greatly expanded scope of applications to which this type of machine, *Fig.* 1, is adapted. The most significant trend was that toward the grinding of contours, typical of which are those in press tools such as lamination and similar dies, *Fig.* 2. These forms include various combinations of radii, tangents, angles, and flats, all of which must be accurately ground to size and location.

To perform these important additional operations, as well as to improve the efficiency of conventional hole grinding, the original jig grinder was redesigned. The new machine, *Fig.* 1, includes some interesting features resulting from the use of inter-related hydraulic, pneumatic, mechanical, and electrical mechanisms. This article presents a design analysis of the machine which, it is hoped, will assist other designers by providing general background and a source of ideas for solving similar problems.

Although the jig grinder is rapidly becoming a recognized basic type of machine tool, it is still new enough to be unfamiliar to many. For this reason it may be well to preface this discussion by stating that, in general, the jig grinder is a machine performing, by grinding, the same functions in hardened materials that the more familiar jig borer does in unhardened material. It is a vertical spindle machine having a horizontal work table capable of being located accurately to rectangular co-ordinate positions in relation to the spindle, by means of precision lead screws.

Hole grinding with the jig grinder is based essentially on the familiar motion encountered in single-point boring; thus, the analogy between jig grinder and jig borer becomes even more apparent. This may be clarified if the grinding wheel is compared to a tool bit held in an outfeed boring chuck, as would be common practice in the case of the jig borer.

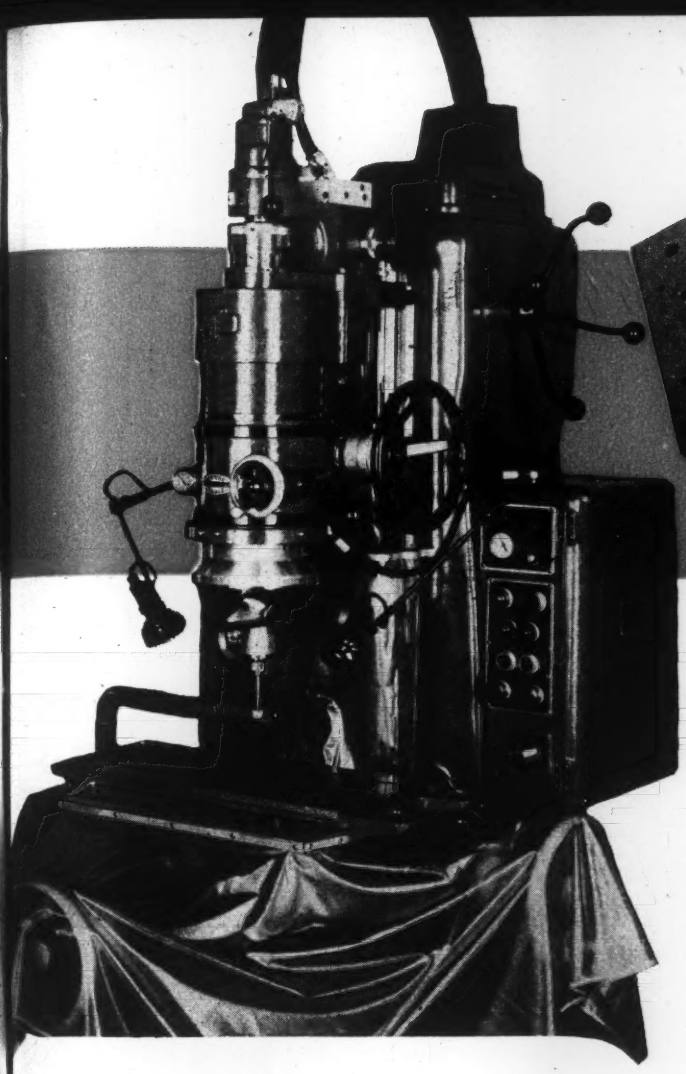
Because hole grinding must be performed with the workpiece and table in a fixed position, it was necessary to impart a planetary motion to the grinding wheel spindle as a means of controlling hole diameter. This was accomplished by vertically mounting the pneumatic turbine grinding spindle on a nominally horizontal, adjustable dovetail slide attached to the lower end of the spindle assembly, *Fig.* 3.

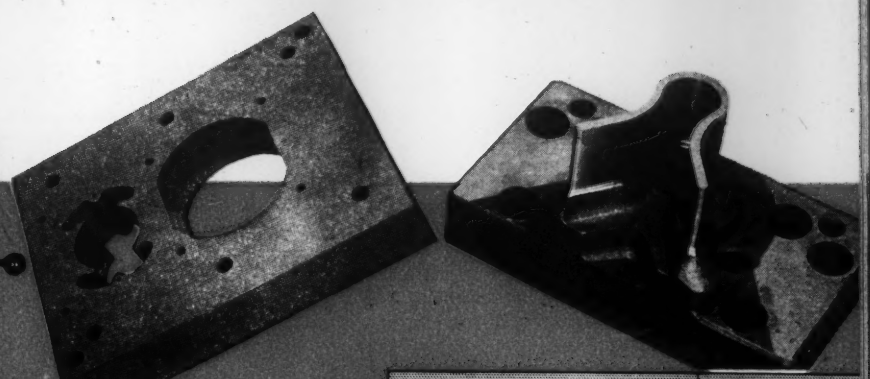
Because the axis of rotation of the main spindle is fixed by a pair of extremely accurate preloaded ball bearings, it is apparent that displacement of the axis of the grinding spindle from that of the main spindle will result in an increase in the diameter of the hole being ground, *Fig.* 3, lower view.

Where Jig Grinder Differs from Borer

Although the preceding description would serve for what might be termed a "basic" jig grinder, it neglects two considerations: the ability to outfeed the wheel accurately while grinding, as well as to grind tapered holes. The importance of these features alone precludes the idea of converting a jig borer to an efficient jig grinder by the mere addition of a grinding spindle to the former machine and, as might be expected, additional refinements and specialization of design have further sharpened the line of demarcation between the two machines.

TAPER SETTING: Because taper grinding resulting from integrating downfeed with outfeed would involve structural as well as operational complications,





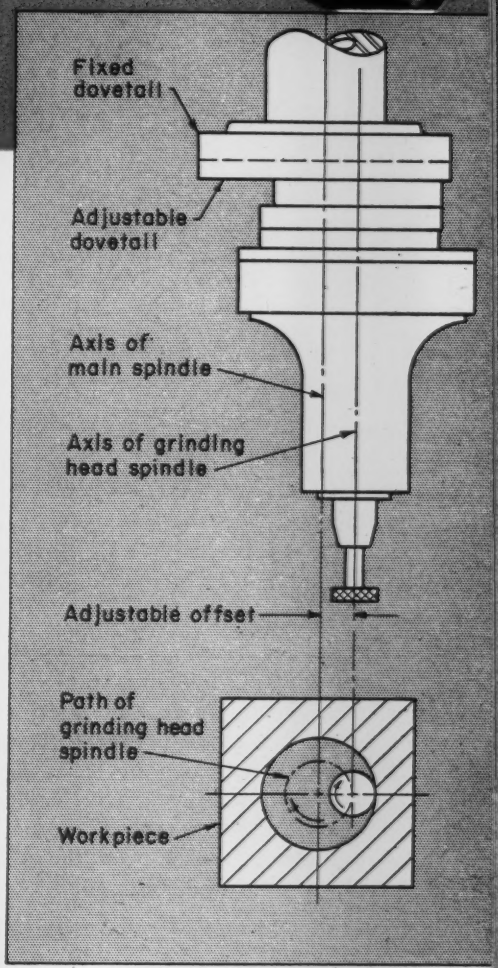


Fig. 1—Left — Moore No. 2 jig grinder designed for precision contour and hole grinding

Fig. 2—Above—Typical workpiece contours ground accurately to size and location on the jig grinder

Fig. 3—Right — High-speed pneumatic grinding spindle adjustably offset from main spindle. Lower view shows planetary motion of grinding wheel around axis of main spindle

and would require the use of taper dressed wheels, a more straightforward solution was sought. This resulted in a spindle assembly, *Fig.* 4, into which an inclinable member was introduced. This inclinable vertical guide sleeve is pivoted at its lower end on a pair of diametrically opposed pivot pins projecting inward through ears on the lower end of the hollow main spindle, *Fig.* 4, center.

Thus, by adjustment of the taper-setting screws, the axis of the bore of the vertical guide sleeve may be inclined at an angle to the axis of rotation of the assembly, *Fig.* 4, right. Likewise, this inclinable axis may be made to coincide with that of the spindle for straight grinding.

In order to transmit the reciprocating motion necessary for grinding, a vertical slide is fitted within the bore of the guide sleeve, *Fig.* 5. This slide is free to move axially in the bore of the sleeve and is keyed to it. The dovetail slide, previously described as carrying the offset grinding spindle, is mounted on the lower end of the vertical slide, *Fig.* 5.

In *Fig.* 5 is also shown the cone generated by the slide as it reciprocates on its inclined axis which is rotating about the fixed, vertical axis of the spindle. By construction, the axis of the grinding spindle remains parallel to that of the vertical slide, whether the latter be set for straight grinding or inclined for taper grinding, and may be outfed in either position, as previously described. It will be observed that even for taper grinding the wheel is dressed to the conventional cylindrical shape, *Fig.* 5.

VERTICAL FEED: Because the basic reciprocating member of the jig grinder is inclinable and located within the rotating spindle assembly, the conventional direct rack and pinion method of imparting vertical movement could not be employed. Consequently it was necessary to introduce a rotating control spindle above the main spindle assembly and coaxial with it. This member could then be reciprocated together with its supporting yoke and transmit both vertical and rotary motion to the upper end of the vertical slide through a tubular, semi-universal drive link. This construction permits the slide to be inclined at an angle to the mutual axes of the main and control spindles, *Fig.* 5.

To maintain alignment between the control spindle and the main spindle, and to permit vertical movement of the former, the aluminum yoke which contains the control spindle is supported on a pair of hardened steel, vertical guide rods. These rods are located at either side and to the rear of the spindle,

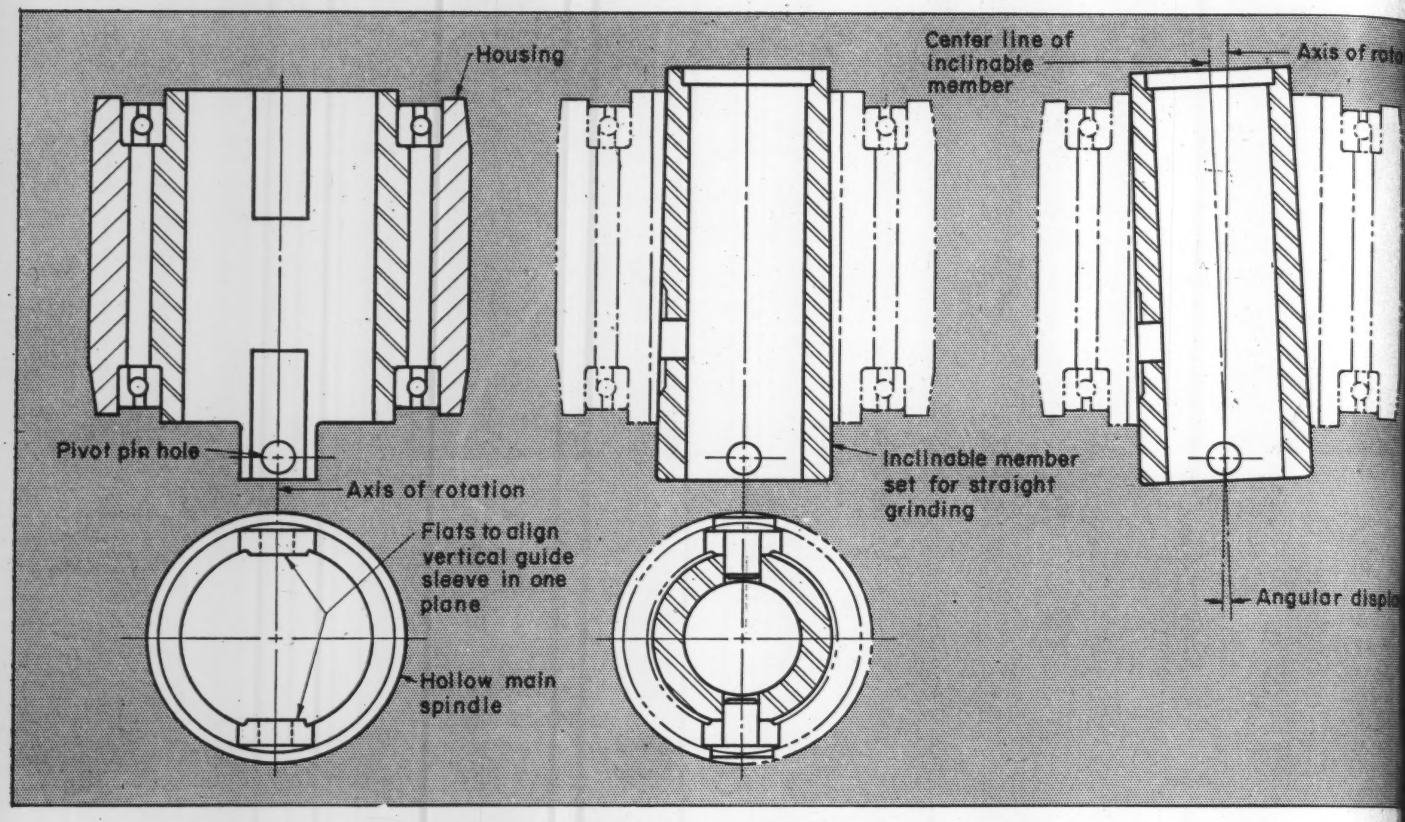


Fig. 4—Main spindle assembly (left), inclinable guide sleeve set for straight grinding (center), and inclinable guide sleeve set for taper grinding (right)

paralleling its axis, *Fig.* 6. The upper ends of these rods are fixed in the yoke, while the lower ends are located, but free to move vertically, within hardened bushings in the main spindle housing.

At this stage of design it was apparent that racks in the lower end of the guide rods, operating with pinions on a transverse shaft through the main spindle housing, would provide a satisfactory vertical actuating means, *Fig.* 6. A handwheel on the end of the shaft serves to complete the mechanism.

OUTFEED: Outfeed while grinding is controlled by means of an internally threaded, graduated dial, much like the barrel of a conventional micrometer, mounted on the nonrotating yoke outside the control spindle and concentric with it, *Figs.* 6 and 7. By virtue of the screw thread, rotation of this dial causes an axial movement which is transferred by an antifriction bearing to a spring-loaded thrust rod system in the rotating control spindle and vertical slide. An inclinable link within the drive link transmits thrust even when the slide is set for taper grinding. A cam on the lower end of this rod, working against a pivoted lever, translates this axial movement into lateral displacement of the dovetail slide carrying the grinding spindle. The dial is calibrated in tenths of one thousandth of an inch, permitting accurate control of hole diameter while grinding, *Fig.* 7.

Coarse adjustment to permit approximate positioning of the wheel in relation to the surface to be ground is effected by means of a fine-pitch screw within the dovetail slide, accessible only when the spindle is stopped. Inasmuch as the thrust from the pivoted lever is applied against a pivoted nut on the coarse adjustment screw, it is a simple matter to spring load the dovetail assembly against this nut. Thus, any backlash is eliminated in both the screw and lever.

SPINDLE HOUSING ADJUSTMENT: In order to accommodate workpieces of varying heights, the main spindle housing assembly is located in vertical double V-ways on the column face and can be positioned within a range of 11 inches. It is counterbalanced by a weight in the column through a roller chain and sprocket system, and may be locked in place during operation of the machine by means of a clamp.

This type of construction avoids excessive overhang of the vertical slide which would magnify any slight inaccuracy in the ball bearings; yet it permits great flexibility of adjustment.

MAIN SPINDLE DRIVE: The preceding basic design features are common to both the original jig grinder and the new model, differing only in minor details of part design. The remainder of the design differs significantly from that of the original, in order to incorporate the desired additional performance features in the machine.

Two apparently irreconcilable factors created a troublesome problem in the design of the main spindle drive. It was necessary to establish the overall, fixed shipping height of the machine at about 90 inches in order to avoid the necessity of horizontal handling in trucks, elevators and doorways. At the same time, in keeping with capacity increases in all other respects, it was mandatory to increase the distance between the work table and the grinding wheel spindle by several inches.

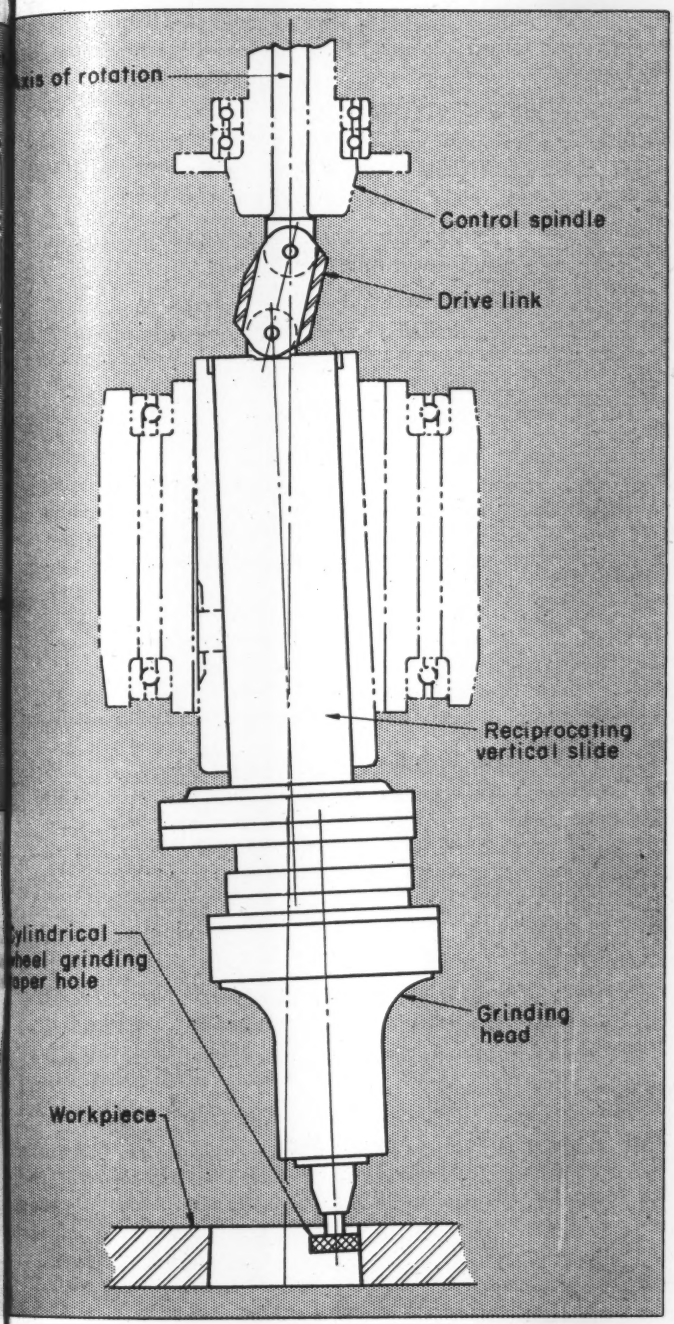


Fig. 5—Reciprocating vertical slide mounted in inclinable sleeve. Control and main spindles are on fixed axis

Requirements of the spindle drive are that it shall provide a range of speeds from 85 to 350 rpm, transmit ¼-hp and permit the spindle to be disconnected from the drive.

Any consideration of a variable-speed drive of the adjustable-pitch sheave type mounted on top of the column was quickly discouraged by the necessary space requirement. Furthermore, experience has proved that, while it is necessary to hold some relationship between main spindle speed and the size of the hole being ground, there is no sacrifice in grinding efficiency over a relatively wide speed range. It therefore became apparent that a four-speed motor would provide an adequate range of speeds and eliminate the complexity of a variable-speed drive.

From the standpoint of simplicity it would have been highly desirable to mount such a motor directly on the yoke, thus eliminating all power transmission problems. This was not possible, however, because such a weight addition could not be tolerated on a rapidly reciprocating member nor could the added height be permitted. Therefore, all other considerations dictated a vertical, shaft-up mounting within the upper end of the column.

The power transmission problem is to connect the shaft of the fixed motor to the control spindle which is vertically adjustable over an 11-inch range and has a 3⅝-inch reciprocating stroke.

Consideration was given to the orthodox systems of V-belt, gear train and fixed shaft types, but in each case mechanical complexity, space requirement and lack of suitability forced a new approach.

Analysis of the actual transmission requirements revealed that, although the motor was rated ¼-hp at 1200 rpm and proportionately at 900, 600 and 450 rpm, the actual power requirements of the machine are but a fraction of that available.

Flexible Shaft Selected for Drive

After thorough test, a specially designed flexible shaft was chosen as the power transmission member. This solution presented several advantages: simplicity, freedom from vibration, minimum space requirement, and ample capacity.

Because a speed reduction of 5 to 1 was necessary between the motor and control spindle, and because a flexible shaft is better suited to transmit power under high-speed, low-torque conditions, it was apparent that the reduction should occur at the spindle end of the drive.

A simple spur-gear reduction was employed. This was housed, together with a positive, tooth type clutch, on the yoke at the upper end of the spindle. This arrangement further served to place the disconnect clutch control within reach of the operator.

As applied, the shaft is looped through a 180-degree arc from the motor shaft to the reduction gear housing on the yoke, *Fig.* 1. The height of this arc is determined by the position of the main spindle housing, so that by lowering the housing it is possible to comply with the shipping height requirement.

POWER RECIPROCATION: Mechanical reciprocation of the yoke and connected member appeared impractical due largely to the necessity for infinite rate from 0 to 120 strokes per minute, and an adjustable stroke from ¼ to 3⅝ inches. On the other hand, the natural identification of reciprocating motion with piston stroke pointed toward utilization of compressed air, already necessary for operation of the pneumatic grinding spindle, and therefore available at 100 psi as a power source.

In its simplest form, such an application would consist of a double-ended cylinder mounted vertically in the yoke, with the piston rod attached to the spindle housing. Reversal of movement could be accomplished by a four-way solenoid valve which would permit pressure to be applied to either end of the cylinder while the opposite end is open to exhaust. Thus, in

effect, the cylinder and yoke would reciprocate on the fixed piston, the stroke being controlled by limit switches and adjustable stops.

Actually, as the design progressed, two such cylinders were found necessary for reasons later to be cited. These cylinders were mounted in the yoke, behind the guide rods.

Experiment showed that operation was satisfactory in a pressure range from 70 to 85 psi, controlled by a pressure regulator and gage.

RECIPROCATION CONTROL: Although compressed air proved to be an ideal power source in most respects, its compressibility resulted in several undesirable traits: nonuniform movement at slow rate of travel, excessive overtravel after reversal of solenoid valve, and difficulty of travel rate control by pressure regulation or metering of flow.

To eliminate the unfavorable characteristics of air as a source of power, while taking advantage of its general suitability, it was necessary to mount a double-ended hydraulic cylinder between the pneumatic cylinders to act as a regulating member. Employment of a double-ended hydraulic cylinder in a closed circuit with an adjustable orifice valve, as a means of controlling air power was not, in itself, unique. However, modifications of that system necessary for this specific application resulted in several unusual design features.

Pressure in the air cylinders, as *Fig.* 8 shows, cannot cause movement of the yoke without an equivalent displacement of fluid from one end of the hydraulic cylinder to the other through the flow-control valve. The fluid, being virtually incompressible, permits infinitely fine control of the rate of travel by means of the adjustable orifice in the valve.

To avoid changing the travel rate setting of this valve to give unrestricted flow for hand feeding, a supplementary by-pass valve was included in the hydraulic circuit. This valve, controlled by a push button through a solenoid, permits instantaneous changeover from auto-feed to hand feed by opening a direct passage from one end of the cylinder to the other.

COMPENSATING FLUID SEAL: Although the hydraulic circuit has been previously referred to as a closed circuit, it is actually only nominally so. In the final version of the design a supplemental, pneumatically pressurized hydraulic sealing system was superimposed on the basic hydraulic control circuit, *Fig.* 8. This solved several problems and offered many advantages:

1. It provided a fluid seal against the cylinder piston rod seals, so that any leakage at that point could not be replaced by air
2. A small (0.0135-inch) passage connecting the control circuit with the sealing system permitted compensation for the volumetric change in the fluid of the otherwise closed circuit. The sealing system, being pneumatically pressurized, permits sufficient bleeding of fluid to maintain a constant pressure in the cylinder regardless of temperature changes
3. The sealing system includes a large reservoir which automatically keeps the circuit full and eliminates the need for any attention to the entire assembly for a period of years
4. Being pressurized, the entire fluid content of the circuit may be easily rid of entrapped air through a bleeder plug at the upper end of the hollow piston rod, *Fig.* 8.

PNEUMATIC COUNTERBALANCE: In the new design, rapid reciprocation and lack of any overhanging mem-

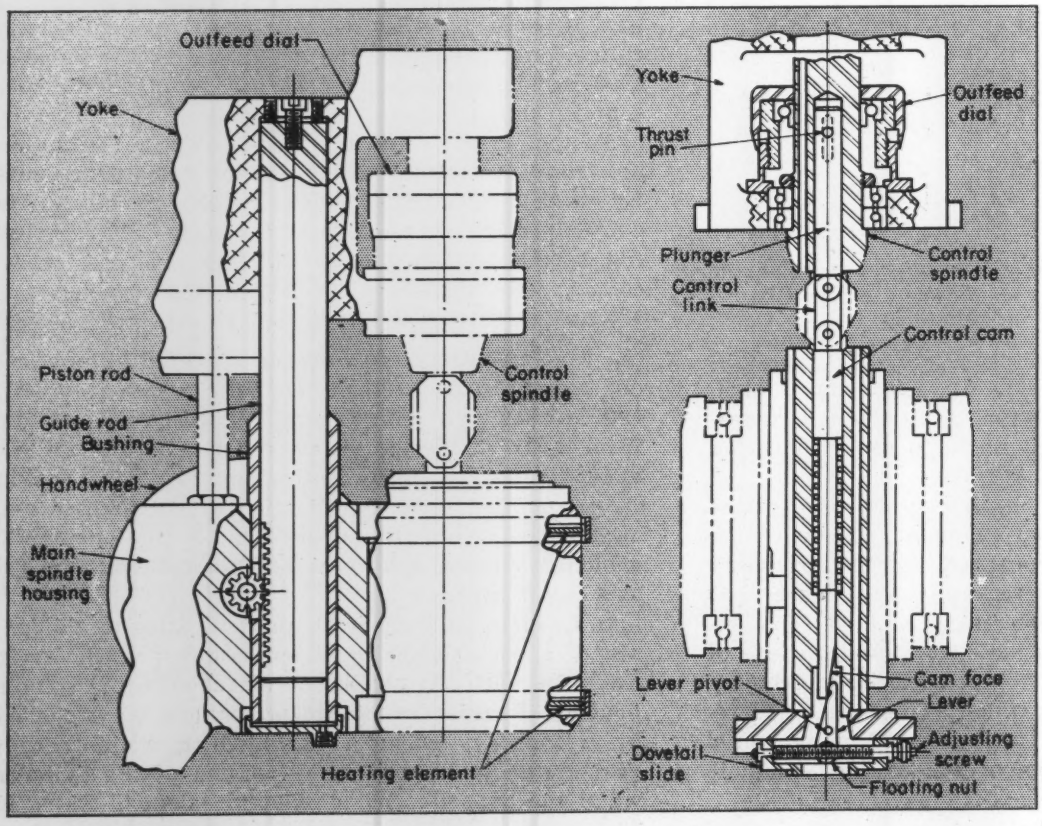


Fig. 6—Extreme Left — Alignment and vertical hand movement of reciprocating members, using guide rods

Fig. 7—Left — Assembly for controlling outfeed while grinding. Dial controls hole size to 0.0001-inch

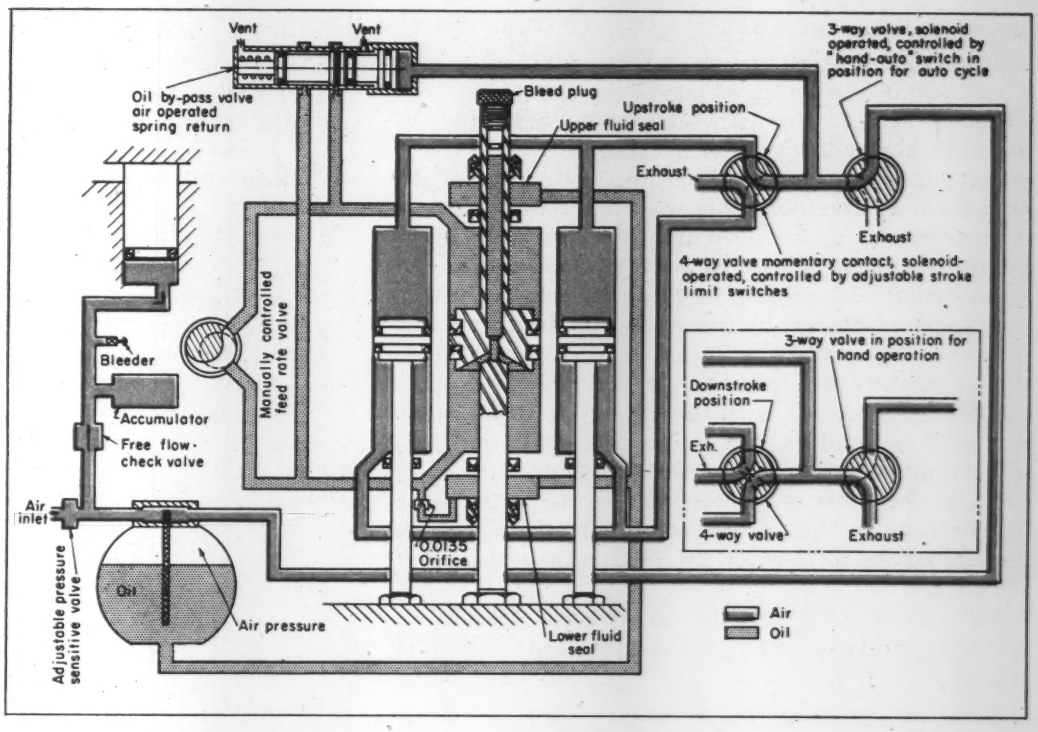


Fig. 8 — Schematic hydraulic-pneumatic circuit diagram

ber from which to support the assembly precluded the use of a counterweight to balance the moving parts.

At first the idea of hollow guide rods enclosing compression springs appeared to present a possible solution. Calculation quickly showed that a four-foot length of spring would be necessary in order to achieve a satisfactorily low spring rate. This scheme was ruled out for lack of space.

Natural similarity of the guide rod and its bushing to a piston and cylinder pointed to their use as such, in order to derive a balancing effect from compressed air.

Application of this idea proved to be relatively simple. A U-cup packing was fitted to the lower end of one of the guide rods and by capping the lower end of the bushing, the chamber thus formed could be pressurized.

Inasmuch at the nominal weight of the parts to be balanced was approximately 73 pounds, and the area of the guide rod end was one square inch, it was apparent that a pressure of 73 psi would support the assembly. One variable entered into the situation—the nonuniform weights of the three different grinding heads and the slot grinder, any one of which might be mounted on the slide. This factor meant that the total weight to be supported might range from 76 to 83 pounds. As previously mentioned, the pneumatic reciprocation cycle would operate without noticeable difference within a pressure range of 70 to 90 psi. The identity of pressure ranges satisfactory for this purpose, and that required for counterbalancing, pointed to the use of the same regulator for both systems. Thus, slight adjustment of the regulator would perfectly balance the weight of the reciprocating parts regardless of which of the attachments might be mounted on the slide, without affecting the power stroke.

As described, this system would serve only as a static counterbalance, while the function of the machine requires the dynamic type. This required a pressure-sensitive valve which would perform the following functions:

1. Admit air to the guide rod bushing as the rod moves upward, holding the pressure constant
2. Maintain constant fixed pressure while the machine is at rest in any position
3. Compensate for volume change as guide rod moves downward, by permitting sufficient air to escape to avoid pressure change.

In its final development, this valve took the form of a self-trimming, line pressure regulator with a compensating relief port on the output side. The resulting system provided a counterbalance so effective that no feeling of weight could be detected during hand operation of the vertical movement.

One further problem was encountered in this portion of the design. When the air supply was shut off overnight, there was no support for the vertically moving parts. To correct this condition a check valve and an accumulator were added to the counterbalance system so that the latter provided the necessary static pressure to support the assembly in spite of failure of line pressure.

OPERATIONAL CONTROLS: Operational controls of the machine may be divided into two classifications: automatic and manual. Because the reciprocating stroke represents the only fully automatic cycle of the machine, it will be discussed first.

Power Stroke, Automatic: The momentary-contact, solenoid-operated, 4-way valve previously described as controlling this cycle is normally energized by two fixed limit switches in the main spindle housing tripped by adjustable stops attached to the yoke, thus causing automatic reversal at either end of the stroke. These stops are so arranged that, by adjusting two pinion knobs operating with rack-cut rods, the stroke length may be adjusted between ¼-inch

and a limit of 3⅝ inches of travel. Any stroke shorter than the maximum can be positioned anywhere within the limits of travel.

Power Stroke, Manual: This cycle may also be manually controlled by a lever arranged to trip either of two limit switches so connected as to cut out the auto-cycle switches. In this way reversal of vertical movement may be instantly and exclusively controlled by the operator. Thus he may reverse the stroke at any point and, so long as the yoke remains within the range established by the auto-cycle limit switches, release of the self-neutralizing lever merely restores the automatic cycle. However, by holding the lever in the desired direction until the auto-cycle limit switch has been over-ridden, the stroke will continue to its travel limit, where it will remain. To return to the auto-cycle stroke, it is only necessary to move the lever momentarily in the opposite direction, thereby causing the stroke to re-enter the zone between the limit switches.

Hand Feed: To use the handwheel for vertical movement it is necessary to complete the following cycle:

1. Cut off the air supply to the pneumatic cylinders and open both ends of the two cylinders to exhaust
2. Move the by-pass valve in the hydraulic control circuit to wide open for free flow of the fluid.

This necessitates a 3-way, solenoid-controlled valve in the line supplying air to the 4-way valve, *Fig.* 8. The function of the 4-way valve being to alternately pressurize the upper and lower ends of the air cylinders and simultaneously exhaust the opposite ends, it follows that one end of each cylinder is always connected to the air supply line while the opposite end is open to exhaust, *Fig.* 8.

The 3-way valve is controlled by two push buttons on the control panel. The button marked "Hand Feed" positions this valve so that it shuts off air to the 4-way valve and opens the previously pressurized port to exhaust, thus completing the first step.

Simultaneously, pressure which has held the by-pass valve closed during power operation is released by the same 3-way valve, *Fig.* 8, permitting its spring to return it to full open position, thus completing the second step.

Instantaneous return to power reciprocation is provided for by a push button marked "Auto-Feed" which reverses the position of the 3-way valve, pressurizing the 4-way valve and by-pass valve for power operation.

Spindle Rotation, Power: Speed selection for the four-speed motor driving the main spindle is governed by a four-position selector switch. Actual starting and stopping at any selected speed is handled by a pair of appropriately marked push buttons.

Spindle Rotation, Hand: The disconnect clutch, mounted on the yoke, is lever operated to permit easy rotation of the spindle. Hand rotation for indicating and job setup may be performed by turning the aluminum guard directly below the main spindle.

Slow, accurately controlled movement for contour grinding may be made by use of a handwheel projecting from the main spindle housing, *Fig.* 9. This handwheel is mounted on a worm shaft which normally is held in a neutral position by a detent plunger. Release of the plunger permits the worm to be moved axially into engagement with a worm gear on the main spindle, thus causing rotation of the entire spindle assembly. An electrical interlock prevents operation of the motor while the worm is engaged.

Angular Arc Setting: In contour grinding it is frequently necessary to grind portions of cylindrical surfaces between accurately determined angular limits. Therefore it must be possible to rotate the spindle assembly back and forth through a controlled arc.

The angle is determined by means of a ring graduated in 5-degree increments, attached to and rotating with the main spindle, immediately below the housing. These graduations are read in degrees against a fixed vernier on the housing.

To avoid the need for continuous reference to these graduations during grinding, the ring was designed with a circular slot of an inverted T shape. Adjustable stops in this slot may be positioned to control the arc of rotation, once it has been determined, against fixed stops in the housing. Thus the spindle may be rotated between positive stops without further reference to the graduations.

To avoid the danger of starting the motor while the stops are in place, the following safety features have been devised. A two-piece circular guard was designed to cover the rotating parts of the assembly. Without the guard in place an electrical interlock renders the motor starting switch inoperative, and the guard cannot be fitted in place until the fixed, angular stops have been removed.

Pneumatic and Electric Spindle Drives

GRINDING SPINDLES: The grinding spindles, mounting on the offset slide, are of two basic types, one powered by compressed air and the other electrically. This is not as contradictory as it might appear; different factors influenced the choice in each case.

Vertical Spindle: Because of the variety of hole sizes encountered in jig grinding, a wide range of spindle speeds is necessary. Fortunately two factors tend to limit the extremes of wheel speed which might otherwise be required:

1. The theoretically ideal speed of 6,000 surface feet per minute for conventional abrasive wheels actually has a rather wide tolerance within which efficient grinding may be performed
2. Experience has shown that for holes below ¼-inch diameter a diamond-charged mandrel is more efficient than a wheel because of the hardness of the abrasive and greater rigidity. In smaller holes, down to 0.015-inch diameter, they are the only available type of abrasive tool. Since these mandrels operate best at a speed approximately one-quarter that of an equivalent wheel, extremely high spindle speeds are not necessary.

As a result of these factors the range of spindle speeds required could be limited to 12,000 to 55,000 rpm.

It would naturally seem desirable to include this

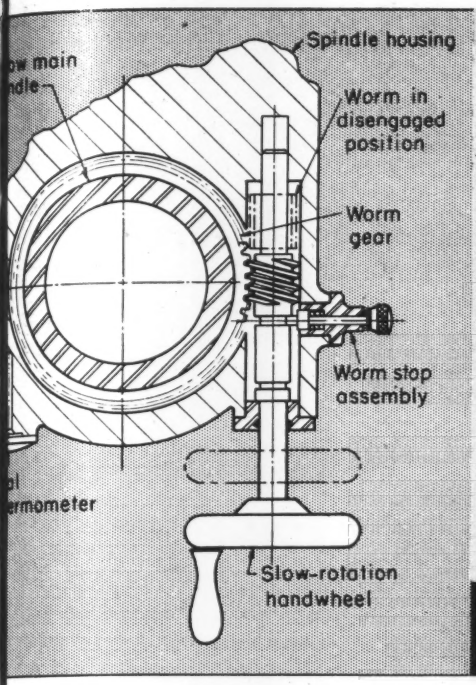


Fig. 9—Worm and gear assembly on main spindle for hand rotation

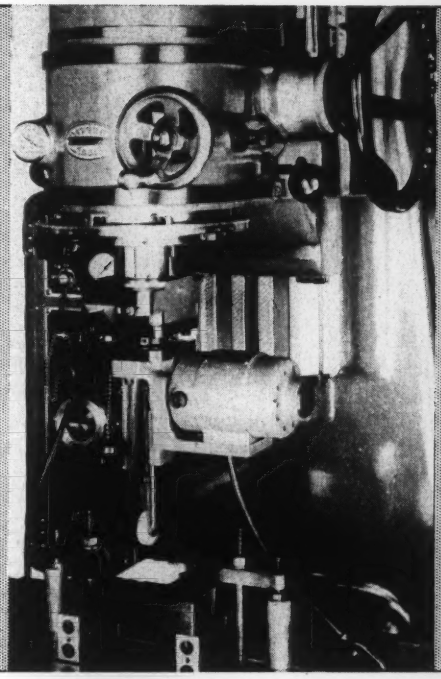


Fig. 10—Typical contour grinding job with horizontal spindle

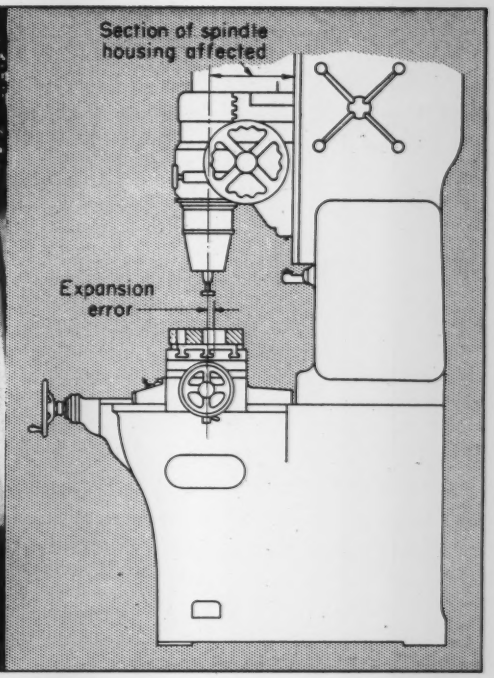


Fig. 11—Effect of temperature variation on accuracy of location

range in one self-powered spindle; unfortunately it is characteristic that, while power is approximately proportional to speed, the requirement here is inversely proportional.

Air, rather than electricity, was chosen as a power source for the vertical spindle for several reasons:

1. Speed can be easily controlled by regulation of air pressure as contrasted with the very complex system of controls and equipment necessary to accomplish this electrically
2. Ample power, up to 1 hp, can be delivered to the spindle in a design much more compact than would be possible with an electric motor of equal power
3. Self-cooling is inherent in pneumatic motors, as a result of the refrigerating effect of volumetric expansion of the air within the motor, thus eliminating the very troublesome problem of heat
4. Load applied, even to the stalling point, cannot possibly damage a pneumatic motor, whereas extensive damage may result from this source in a high-speed electric motor
5. Only one connection is necessary in a pneumatic motor, the air supply, whereas, in addition to the input and exhaust lines needed for the cooling medium, electric current must be conducted to the planetarily rotating head through slip ring or similar troublesome collectors.

Three models of the vertical spindle type were necessary to cover effectively the speed range required, without excessive power loss.

For low speeds (10,000-14,000 rpm), a positive displacement, vane type motor was chosen for its ability to develop power at low speed.

The range from 20,000-40,000 rpm, as well as that from 40,000-55,000 rpm was covered by two models of the impulse-reaction turbine type, differing essentially only in rotor diameter. This design avoided the problem of friction at high speeds, which would be excessive in the displacement type due to centrifugal loading of the sliding vanes.

Although rapid interchange of the various spindles is provided for by a quick-acting mounting on the slide, it was found that the versatility of the 20,000-40,000 rpm head permitted it to be used efficiently for a range of holes from 0.030 to 3 inches in diameter, thus taking care of about 90 per cent of all normal requirements.

Horizontal Spindle: The horizontal spindle presented an entirely different set of conditions affecting the choice of power source. Essentially this accessory was designed to permit grinding of contours consisting of flat surfaces, either straight or angular, or relatively large radii, within the workpiece. A typical example is shown in *Fig.* 10.

An electric motor drive was chosen after careful consideration of the following factors:

1. Due to the nature of its function, the horizontal spindle is never rotated through an arc of more than 359 degrees; thus a conductor may be directly connected from power source to motor
2. With the motor mounted horizontally above the spindle, a belt drive permits a speed increase from motor to spindle by a ratio of pulley diameters, and with a 1½-inch diameter wheel, a standard 18,000-rpm motor could be used
3. Heat in this type of motor is slight and is dissipated by an integral fan in such a direction that no heat reaches the machine or workpiece.

STABILITY CONTROL: Because of the extreme operational accuracy requirement of this type of machine, the problem of dimensional stability becomes extremely important. Common sources of instability include:

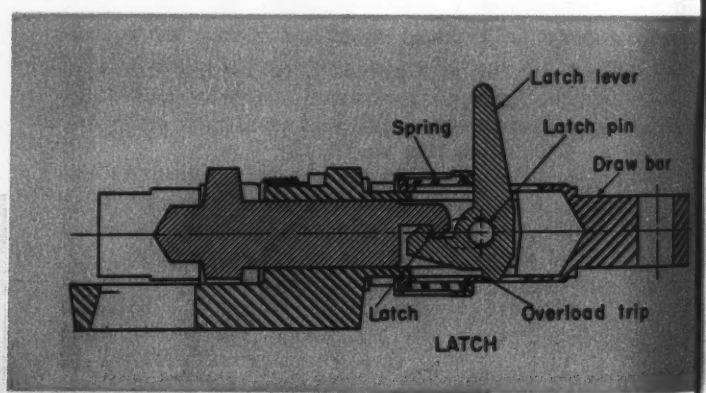
1. Deflection of structure resulting from stresses

(*Concluded on Page* 204)

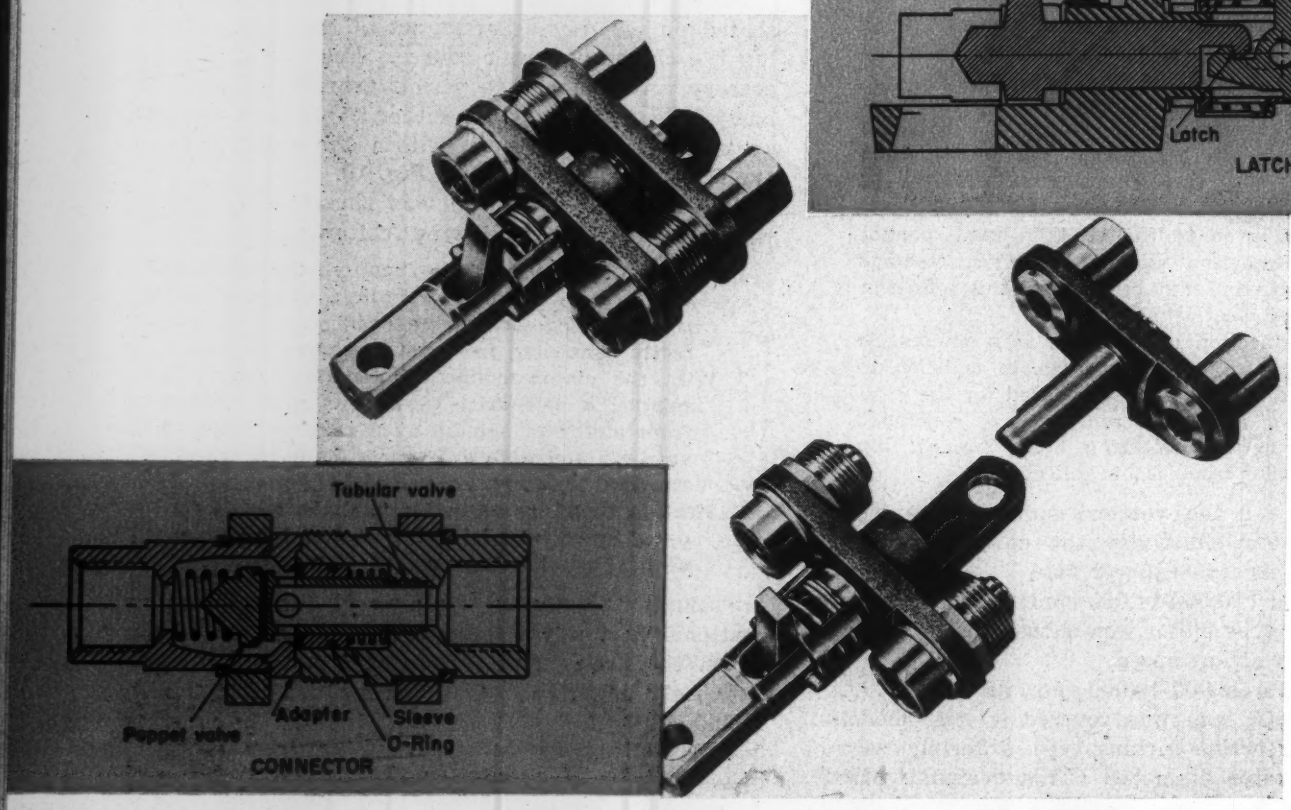
SCANNING the Field For Ideas

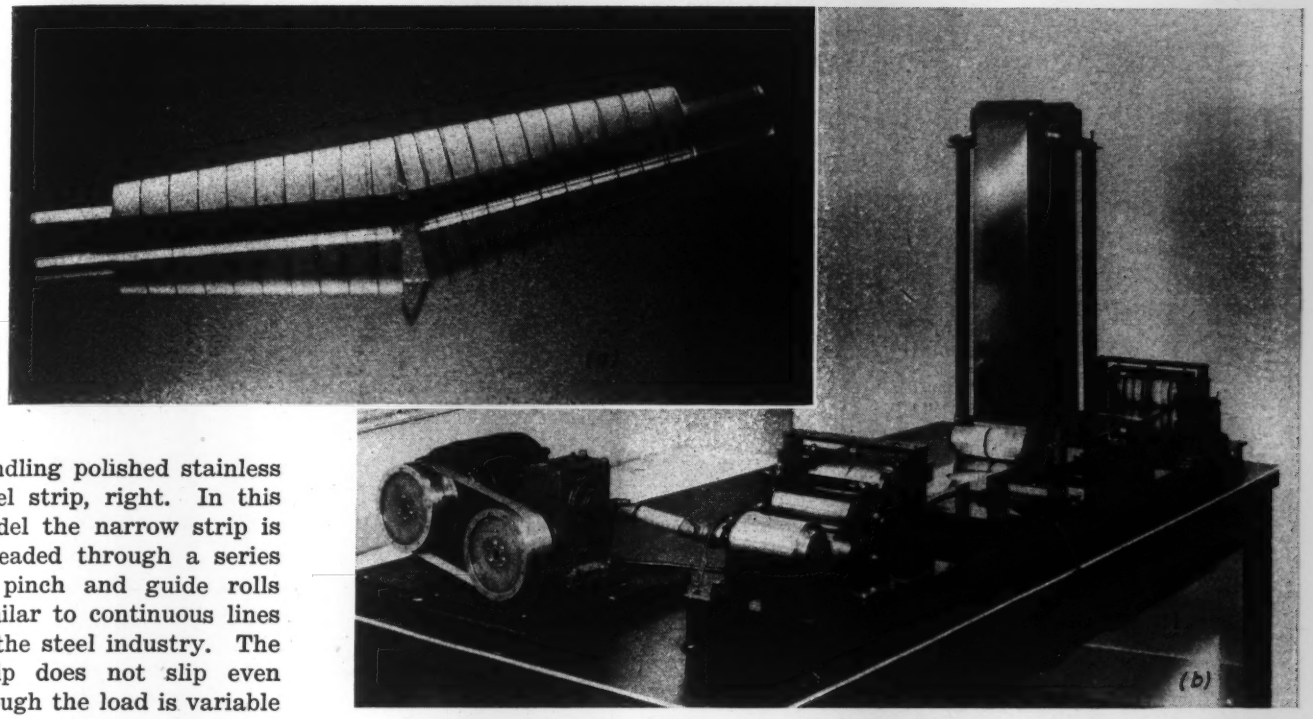
Quick-release coupling, below, for hydraulic lines between tractor and trailer or where disconnect is desirable under hydraulic pressure has been developed to protect hose lines against damage and loss of hydraulic fluid. Designed by Aeroquip Corp., the coupling utilizes two self-sealing connectors and a mechanical latch. Shown assembled and disconnected, the coupling may be separated automatically by an external tug on the hose lines or manually by moving the latch lever. Features of each unit in the coupling are shown in cross-section. When assembled, both valves in the connector are held open as shown. The tubular valve stem of the right-hand member holds the poppet valve open and the adapter on the left-hand member holds the valve sleeve open. When disconnected, both of the valves close with practically no loss of fluid.

The latch closes readily when the two halves of the coupling are assembled and is held in the latched position by a spring. Manual release compresses the spring, unlatching the coupling. Overloads compress the spring and pivot the trip about the latch pin until the latch opens, separating the coupling.



Self-centering rolls, employing the principle of "planar action", do not require side guides for conveying materials as illustrated by the model mill for





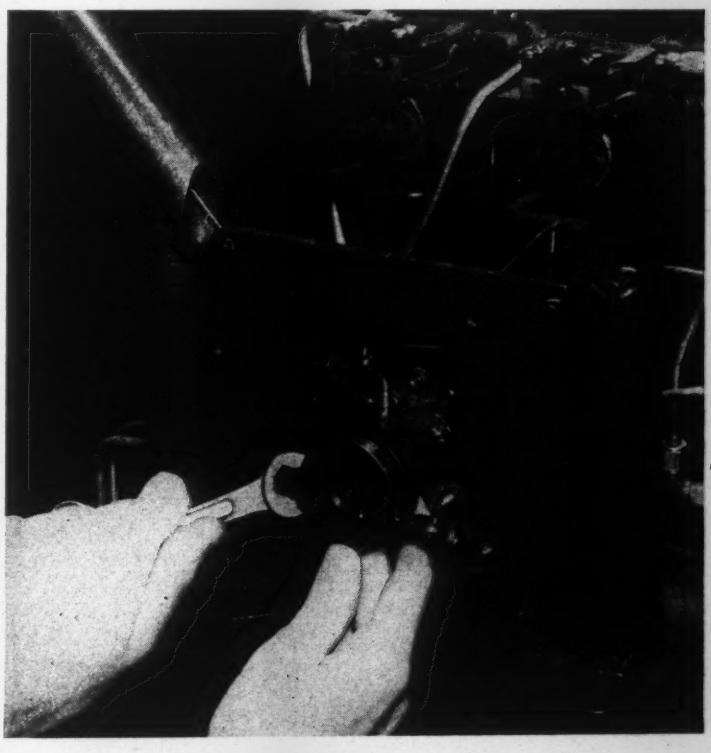
handling polished stainless steel strip, right. In this model the narrow strip is threaded through a series of pinch and guide rolls similar to continuous lines in the steel industry. The strip does not slip even though the load is variable and the drive and take-off are overhung.

Invented by E. T. Lorig, chief of the senior engineering staff of Carnegie-Illinois Steel Corp., the self-centering roll is a slightly crowned roll cut transversely at the center. The two halves are fixed to rotate as a unit so that the working surfaces of the two parts are approximately horizontal while the axes are at an angle as shown in the Lucite model at top. In this way the planes, or lines of force, in both halves twist evenly toward the center in the direction of movement to confer a self-centering action on any materials passing over the rolls. Many of the mechanical problems involved in design were solved by William Mursch, an engineer on the senior engineering staff.

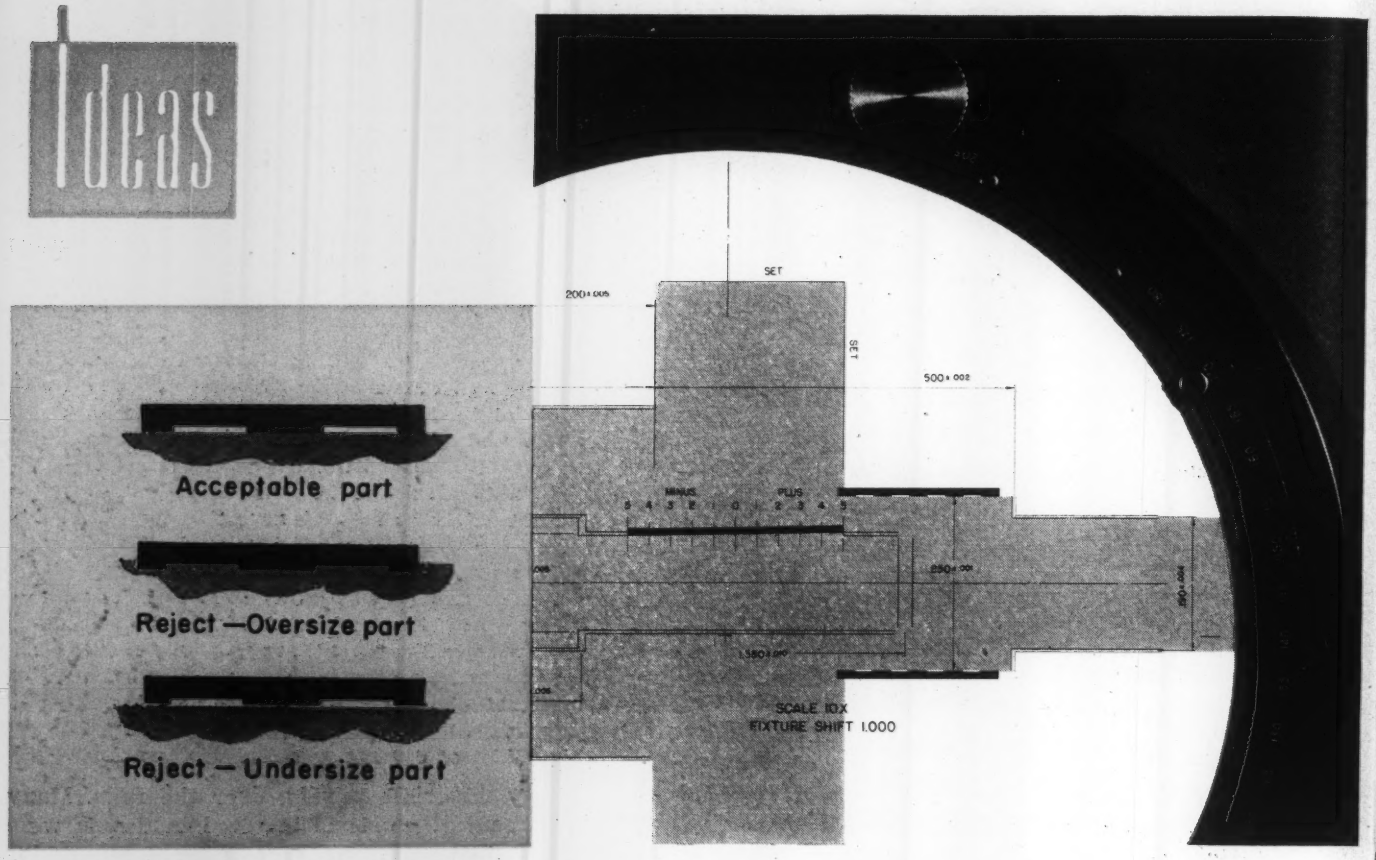
The rolls have been tested for applications such as runout tables, feed and guide rolls for electrolytic tinning, galvanizing and pickling lines, gravity conveyors, and belt pulleys. When arranged to form a roller conveyor, the rolls retain their self-centering action even around a curve in the conveyor.

Scale-model testing of a small rocket motor, right, is providing much needed information on fuels and their behavior under low temperature and pressure conditions. This tiny motor runs on small amounts of fuel and eliminates the need for elaborate safety precautions, making possible experiments costing far less than would those with full-scale models. A Navy sponsored program at the Massachusetts Institute of Technology, the studies have proved their value in testing fuel efficiencies and in gaining a more complete understanding of fuel combustion. Future work is expected to provide answers to specific rocket design problems.

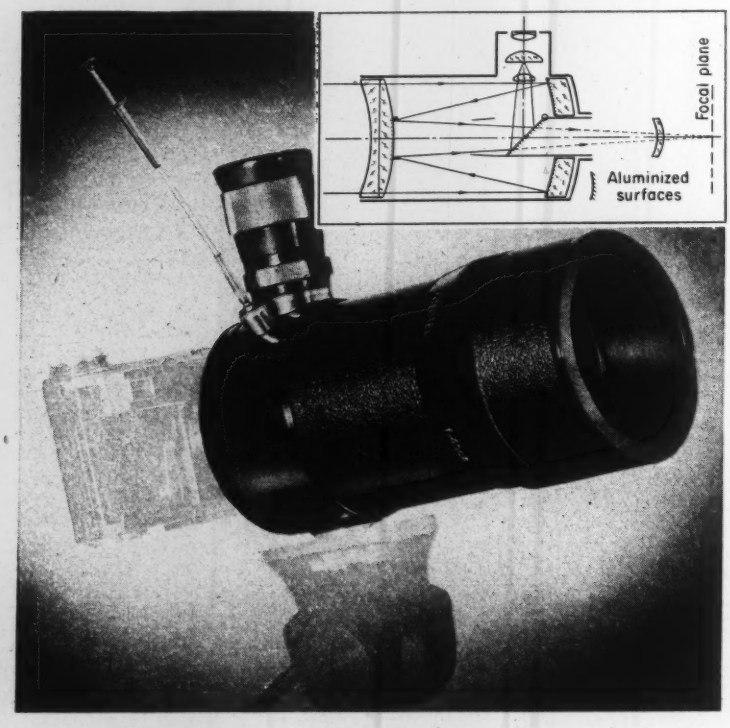
This microrocket uses only two pounds of fuel in one minute, whereas, during the same period, 12 tons would be consumed in a rocket the size of a German V-2. Jets of the microrocket fuels, shot through 0.01-inch holes, come together inside the chamber which is ½-inch in diameter. More than 300 psi pressure is built up inside the motor and the nozzle velocity of the gases is about 5000 miles per hour, more than 2½ times the velocity of sound at these conditions.



Ideas



Accurate gaging to split-thousandths for go and no-go limits is facilitated on optical comparators by use of screens, above, having micro-gage bridges. Made by Engineers Specialties Division, these bridges are highly legible and dependable within the gaging limits. Light must be seen within the bridge on the screen if the part is within permissible tolerances. The contrast of light and shadow is amplified by the bridge as illustrated in the sketch. For an acceptable part, light must be seen within the indentations but not below the footing of the bridge. Deviations beyond acceptable tolerances are thus easily seen.

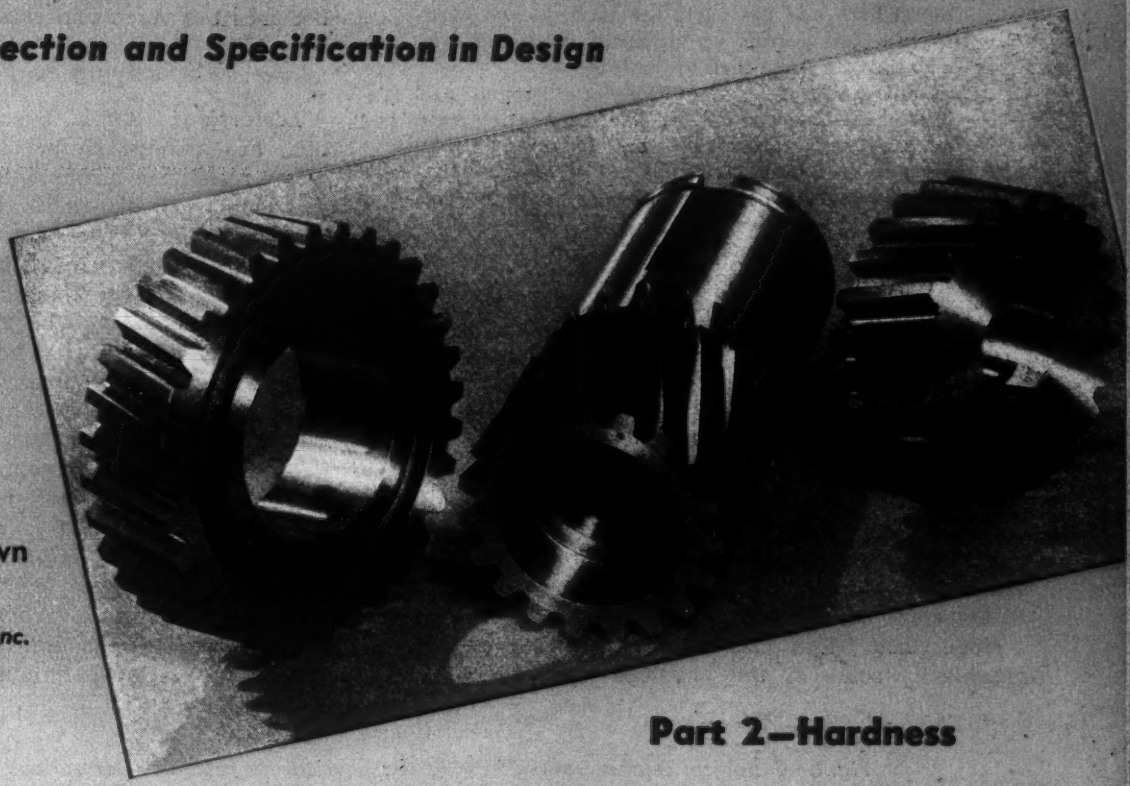


Compact telephoto lens, left, employs the principle of reflecting light rays within the lens system to increase effective focal length. Developed in Holland and distributed by Burke and James Inc., the lens "folds" its focal length back and forth three times by reflecting the light rays upon two mirrors—one concave and the other convex—and then through a correcting lens. Having an 18-inch focal length and a speed of f:56, the unit is six inches long, including a reflex focusing device as shown in the sketch of the optical system.

Light rays traveling through the concave-convex lens are reflected by the concave aluminized-surface mirror back to the central area of the first lens which also is a mirror surface. The secondary reflection travels through a hole in the center of the concave mirror and through the lens. A reflex mirror, in front of the concave mirror, projects an image through an eyepiece for viewing. The reflex mirror flips out of the way during exposure of the film and returns after a picture is made.

HEAT TREATMENTS

. . . Their Selection and Specification in Design



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Part 2—Hardness

DEVELOPING the study of heat treatment specification according to the three major physical property divisions outlined in the first article —toughness, hardness and combination hardness and toughness—this part covers hardness and the pertinent factors affecting design specifications. Hardness can be considered in a general sense as resistance to wear and indentation. Treatments which can be employed to cope with this requirement in design are covered in the following paragraphs. Normalizing as related to hardness is principally used in the processing of parts that have to be wear resistant and is an auxiliary to one of the other treatments.

CASE HARDENING: This involves the use of a low-carbon steel (under 0.25 per cent carbon—Groups I and II, *Fig.* 1) or, on occasion, a medium-carbon steel and increasing the carbon content near the surface which is hardened in subsequent processing. In the interests of hardness alone it is only necessary to use a plain-carbon steel. The alloy grades in Group II, *Fig.* 1, should be considered when distortion is a serious problem and oil quenching is indicated instead of water quenching; the alloys also tend to get a harder and more uniform case than the plain carbon steels in larger sections, over 3 or 4 inches in thickness.

Specifications Should Be Flexible

Much has been wrtten about the various methods of case hardening, especially pack, gas and liquid processes. Our interest in them is confined more to the type of case produced (toughness, brittleness, depth, etc.) than to the actual mechanics of the treatment. For instance, if a desired case can be obtained by more than one method then it is to everyone's interest to leave the method flexible in the specifications so advantage can be taken of the means available, whether that be in the company's plant or at a commercial steel-treating plant. Of course, when conditions warrant, definite methods and equipment are set up—as in automotive plants—and in such

cases plant metallurgists and other personnel are called into action.

Case hardening is usually accomplished by one of four ways:

1. *Carburizing:* Carbon is added to the surface and the parts are then quench hardened. Case depths up to about 0.070-inch are obtainable
2. *Cyaniding:* A combination of nitrogen and carbon is added to the surface by heating the part in a liquid bath and the parts are then quench hardened. Case depths up to about 0.010-inch and occasionally 0.015-inch are obtainable
3. *Carbonitriding:* A combination of nitrogen and carbon is added to the surface by heating in a suitable gas atmosphere, and parts are then quenched. Distortion is usually less than that experienced by carburizing or cyaniding. Case depths up to about 0.030-inch are obtainable
4. *Nitriding:* Nitrogen is added to the surface by heating in a suitable gas atmosphere and holding there at the proper heat for an extended period of time (2 to 4 days) and cooling without quenching. Special-analysis steels under the name Nitralloy are most commonly used for nitrided parts although many medium-carbon alloy steels such as AISI A-4145, A-6145, etc., can be employed. Case depths up to about 0.045-inch are obtainable with 0.010 to 0.015-inch being usual for the higher range of hardnesses.

The hardness and wear-resistant properties of the nitrided case are best, the cyanided and carbonitrided cases are next best, while the carburized case runs a close third. The chart shown in *Fig.* 8 portrays graphically some of the physical characteristics of the various types of cases as related to each other.

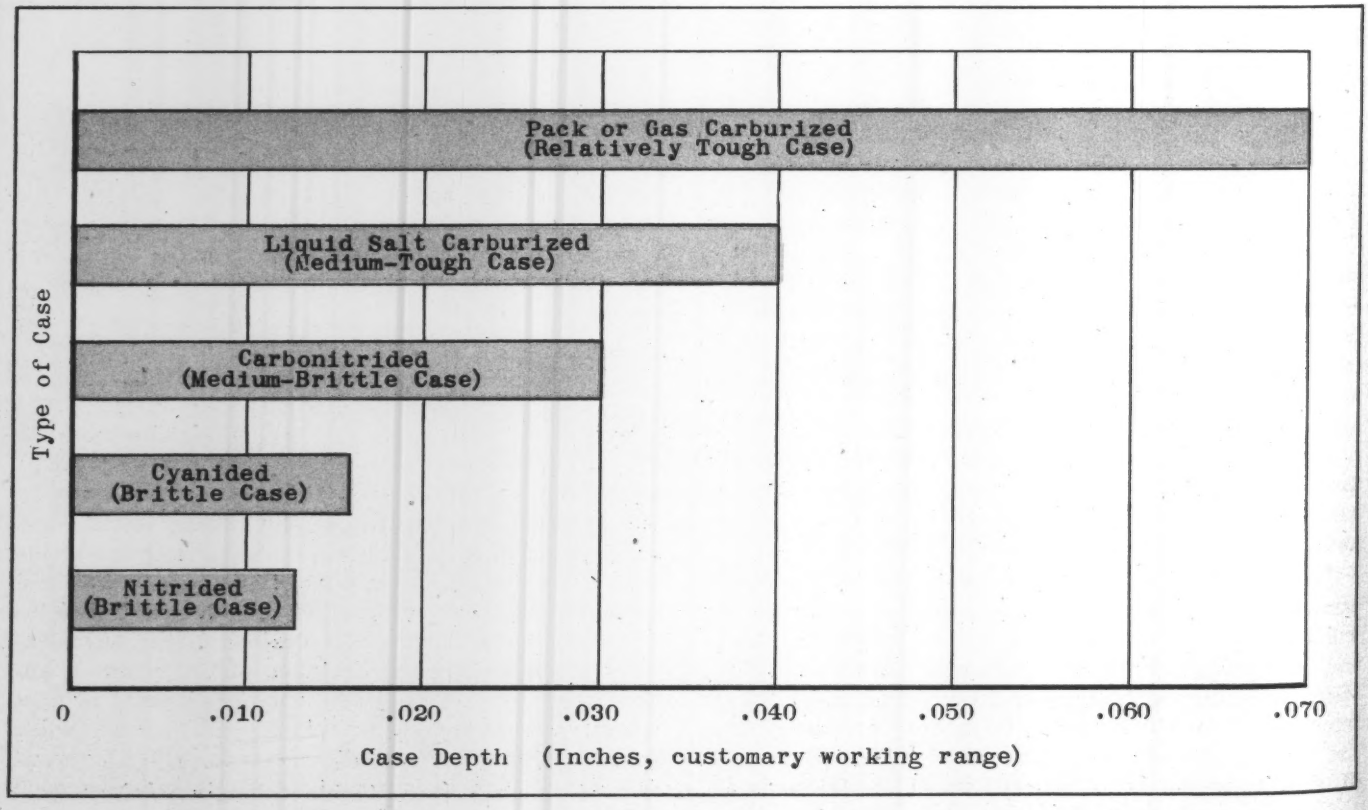
Applications in which wear resistance is of most importance—where stresses such as excessive pressure and shock are negligible—desirable depth of case will generally depend upon the amount of distortion that might occur in treating and the amount of grinding required, which will remove some of the case. The depth specified also must depend upon the amount of wear permissible in service before the part becomes too much undersized for further use.

Specifying to Prevent Warpage

Distortion: Axles, bushings, rollers, gears, pins, etc., are among the parts fitting into this strictly wear resistance group. The conditions mentioned previously will determine the minimum depth of case required. Unlike the relatively fixed factors of grinding and wear allowance, the amount of distortion to anticipate requires serious consideration. The best procedure appears to be that of suggesting various processing methods that will reduce as much as possible the amount of distortion that might reasonably be expected during case hardening of various parts. General design of the part will aid in setting up reasonable distortion grouping.

Negligible Distortion: This group can be applied to parts that are of regular, chunky design such as certain type guides, latches, etc. Those of small and medium cross section (up to about 3 inches) can be made from low plain-carbon steels in Group I, *Fig.* 1, while those of large dimensions are generally better made from a low-carbon alloy in Group II, *Fig.* 1.

Fig. 8—Graphical representation showing the physical characteristics of various types of hardened cases. Data are intended for use as a guide inasmuch as conditions may alter the limits



The reason for this is that the plain-carbon steels do not respond as well to treatment in the larger sections and frequently the resulting case is neither uniform nor up to full hardness. The alloys, however, are more dependable in this respect. The drawing specification should read:

Machine
Carburize
Harden (60 to 65 Rockwell C)
Grind

Low Distortion: This applies to a wide variety of parts such as bolts, nuts, bushings, gears, etc. Up to about 3 inches cross section, parts in this bracket can be made from plain low-carbon steels with the following drawing specification:

Rough machine
Normalize
Finish machine
Carburize and harden
Grind

Low-carbon alloy steels can be used in all sizes and the drawing specification can read:

Machine
Carburize and harden
Grind

Medium Distortion: Axles, shafts, ball races, bushings, gears, mandrels, pinions, etc., fall into this class. If facilities for carbonitriding are available, it is possible that a plain-carbon steel in Group I, *Fig.* 1, could be used. The drawing specification should read:

Machine
Carbonitride
Grind

However, the alloy grades in Group II, *Fig.* 1, are most commonly used. With a normal amount of machine work the drawing specification should read:

Machine
Stress relieve
Carburize and harden
Grind

If an exceptional amount of machining is involved on the low-carbon alloy then the specification should read:

Rough machine
Normalize
Finish machine
Carburize and harden
Grind

Considerable Distortion: Slender parts, large thin rings and parts of complicated or irregular shape are all likely to fall into this category. Low-carbon alloy steels are usually used on these jobs when resorting to carburizing. Practical specification is as follows:

Machine
Carbonitride
Grind

If adequate time is available and the part warrants extra careful processing then the foregoing specification could be expanded to read:

Rough machine
Normalize
Finish machine
Carbonitride
Grind

Where carbonitriding facilities are not available, alternate specifications are:

Rough machine
Normalize
Finish machine
Carburize
Martemper (instead of quenching in oil in the usual manner)
Grind

or

Rough machine
Carburize
Straighten
Finish machine
Stress relieve
Martemper
Grind

If a thin case is satisfactory and the time and cost of the treatment is justified, then nitriding can be considered. In this instance one of the nitriding steels such as Nitralloy or suitable medium-carbon alloy would be used and the drawing specification would read:

Machine
Stress relieve
Nitride

The foregoing suggestions for specifying case hardening treatments can be varied according to the needs of the part, allowable cost, etc. For instance, stress relieving is frequently used instead of normalizing. Also, selective case hardening is often possible by blocking off portions of the part not wanted hard by copper plating or other means. Another method is to leave the portions to be soft oversize while machining the other portions to size. Machining the oversized portions to size removes the carburized case and the sections remain soft in the subsequent hardening operation.

Fig. 9 is an outline form of the case hardening specifications discussed. It will be noted that it is confined to holding distortion to a minimum and no attempt has been made to specify other physical case requirements such as type, depth, etc.

SURFACE HARDENING: When high hardnesses over 60 Rockwell C are not necessary and 55 to 60 Rockwell C is satisfactory, then flame or induction hardening can be used. An advantage of these methods is that distortion is very low on symmetrical parts and the treatment is fast. As far as wear resistance is concerned it is only necessary to use a plain 0.40 to 0.45 per cent carbon steel such as AISI C-1045.

When wide surfaces are to be flame hardened, caution should be exercised before making up the specification to see that the shop or treating plant has the proper equipment to do a good job, especially if a surface of uniform hardness is required. Some shops have flame jets only 4 to 6 inches wide which

CASE HARDENING SPECIFICATIONS

Type Steels	Anticipated Distortion According To Design Features			
	Negligible Distortion	Low Distortion	Medium Distortion	Considerable Distortion
Low, plain-carbon C-1020 B-1112 C-1117, etc. (All Sizes)			Machine Carbonitride Grind	
Low, plain-carbon C-1020 B-1112 C-1117, etc. (Under 3 in.)	Machine Carburize (Or Cyanide) Harden Grind	Rough Machine Normalize Finish Machine Carburize (Or Cyanide) Harden Grind		
Low-carbon alloys A-4615 A-3115 A-8620 A-5120, etc.	Machine Carburize (Or Cyanide) Harden Grind	Machine Carburize (Or Cyanide) Harden Grind	Machine Stress Relieve Carburize (Or Cyanide) Harden Grind * Rough Machine Normalize Finish Machine Carburize (Or Cyanide) Harden Grind	Machine Carbonitride Grind * Rough Machine Normalize Finish Machine Carbonitride Grind * Rough Machine Normalize Finish Machine Carburize Martemper Grind * Rough Machine Carburize Straighten Finish Machine Stress Relieve Martemper
Medium-carbon alloys A-4140 A-4640, etc. Nitralloy				Preheat-treat (If Necessary) Machine Stress Relieve Nitride

Fig. 9—Outline form of case hardening specifications. Treatments are arranged to permit lowest production cost compatible with the minimum practicable distortion in hardening. Parts subject to considerable distortion owing to intricacy of design features naturally result in maximum treating cost to prevent warpage

travel across the surface being processed followed closely by an aqueous quenching spray which results in the surface hardening. Therefore, when the surface is wider than the flame jet it is necessary to re-

peat the operation in a path next to the portion just treated and to continue the procedure until the entire surface is covered. Even with the best of care it is possible that the jet, as it travels next to the previously hardened band, might tend to draw or temper the edge of that band to a lower hardness to the extent that the finished surface would have a streaky hardness. Cylindrical parts such as rolls and shafts, however, can be hardened satisfactorily by progressive spinning.

Flame hardening has been and still is well adapted to small-lot production. With new flame hardening equipment on the market it has also invaded the production field in which induction hardening has been prominent. Induction hardening is similar to flame hardening in that in most instances the surface is hardened by heating rapidly and quenching. Surface hardening specification should read:

Machine
Stress relieve (if part is of intricate design or much machine work is involved)
Flame or induction harden designated surfaces

THROUGH-HARDENING: Plain, high-carbon steels (over 0.60 per cent carbon—usually AISI C-1080 to C-1095) and the water and oil-hardening alloy and tool steels are sometimes used on wear resistant parts when it is felt the high hardness can be obtained with less distortion or with less effort and cost than by using a case-hardening treatment. They are also used when greater depth of hardness can be used to advantage. Obviously, the water-quenching steels are used on short, symmetrical shapes while the oil-quenching grades are used on the more intricate designs. Mandrels, pins, shafts and rolls are among the parts that might utilize this process.

Applications of this through-hardening method in the interests of hardness alone are limited and it is usually best to ask the advice of the steel company or local heat treater before deciding definitely to go ahead with this setup, assuming that there has been no prior experience with the part involved. Factors which might influence consideration are that the high-carbon steel might be available in the plant and no low-carbon in stock; or the heat-treating department in the plant might be more familiar with through-hardening methods and not have any case-hardening equipment. The usual specification reads:

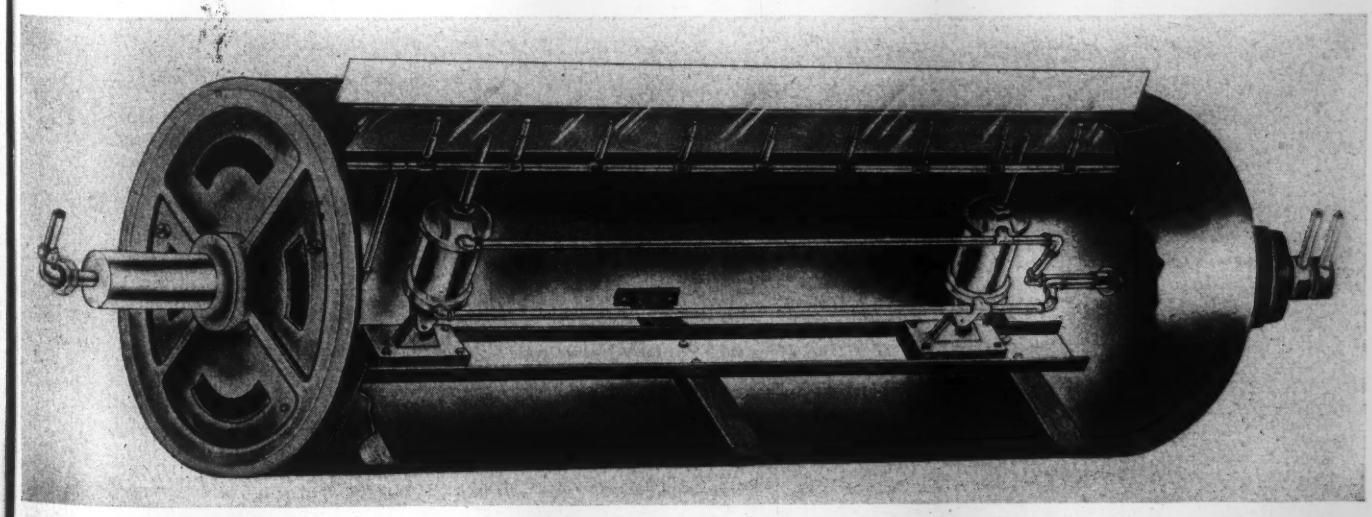
Machine
Stress relieve
Heat treat
Grind

Distortion of parts in sections under 3 inches can be reduced by Martempering, which has been mentioned previously. The treatment is most effective on the oil-quenching grades of steel, although occasionally smaller sections of water-hardening steels are handled in this manner.

The third and last physical property of interest from the designer's viewpoint is combination hardness and toughness, and will be covered in the concluding article, Part 3 of this series.

High-Speed Cut-off Is Pneumatically Operated

UNIQUE drive of an automatic cut-off knife on wet board paper machines uses pneumatic cylinders contained within the cut-off roll. The cylinders are electrically controlled through relays and an adjustable electric counter which is set for the number of roll revolutions to give the desired sheet thickness. Designed by the Noble & Wood Machine Co., the roll uses air jets timed with the knife stroke to help strip the sheet from the roll after a cut is made. New plies for the next sheet start building up on the roll immediately. Since action of the knife is instantaneous, the roll need not be slowed down for the cut-off operation, with resultant production increase.

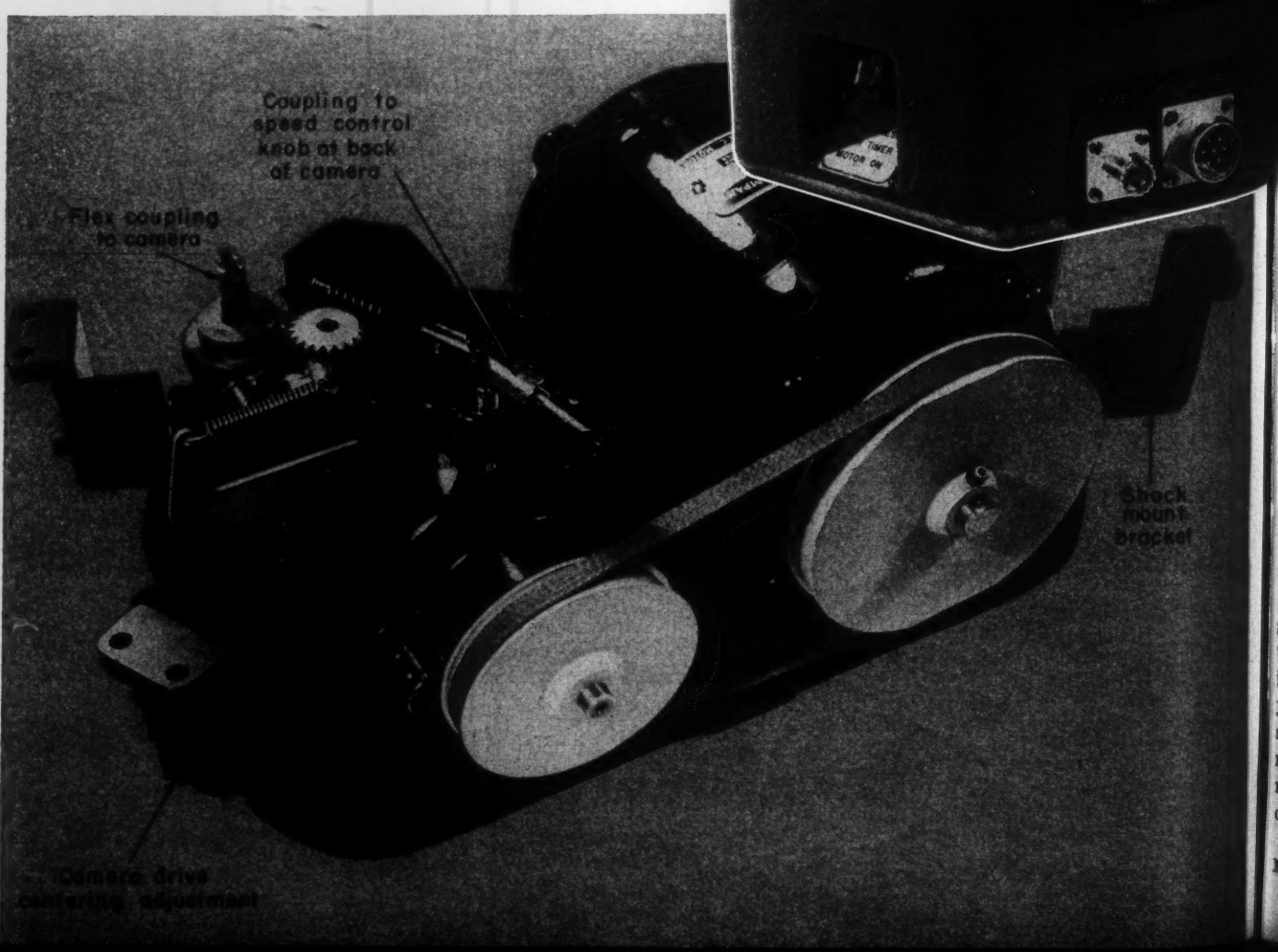


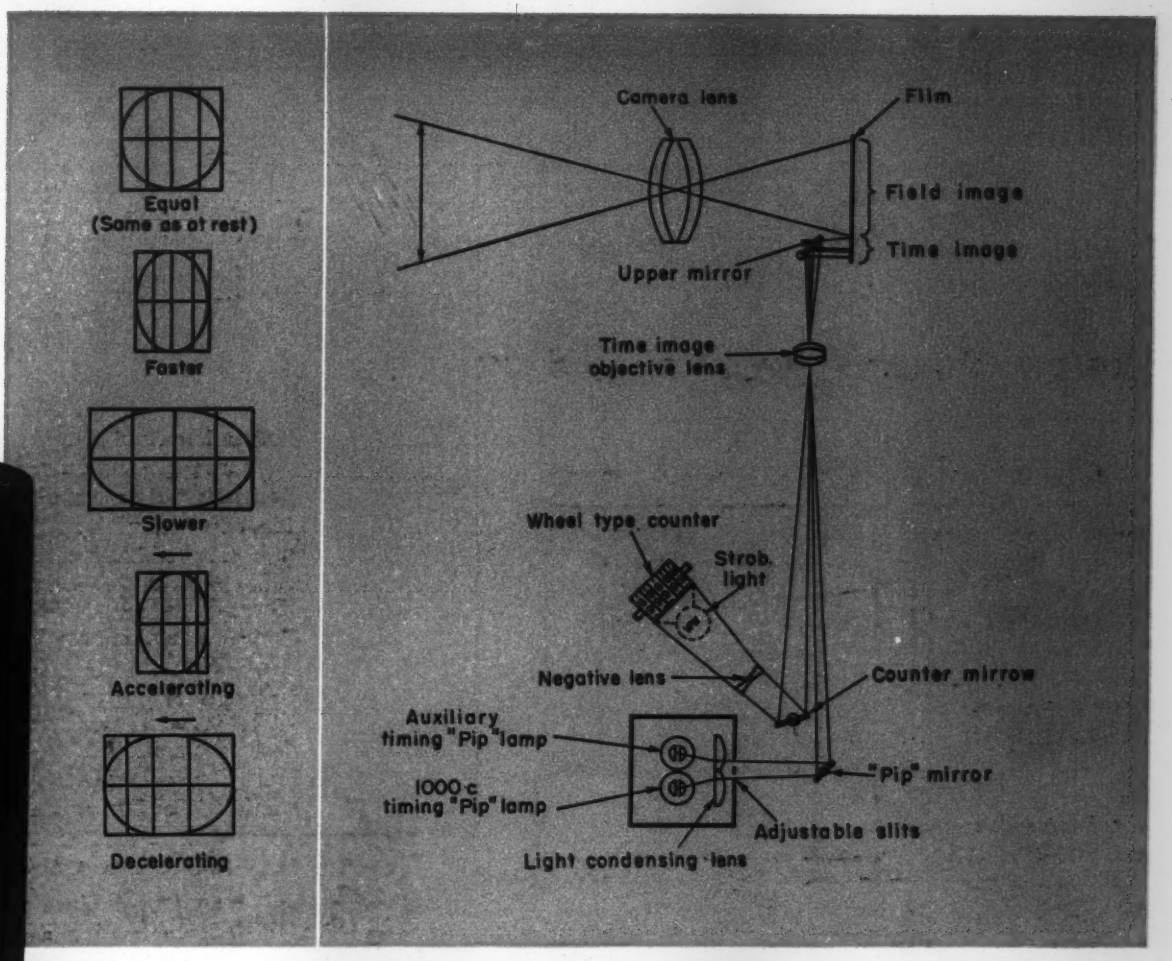
Contemporary DESIGN

Camera Design Based on Slit Principle

VELOCITY and acceleration of objects moving at 2000 miles per hour can be recorded and measured, using a single optical image, with the Beckman & Whitley slit type camera. A slit camera, shown at the right, records the time sequence of events passing or remaining within a vertical plane through the optical axis of the lens. In addition to photographing the event, the machine automatically records, on the same film, electronically timed numbers showing elapsed seconds and hundredths with intermediate pips spaced at thousandths of a second. The camera can also be used to make conventional time-and-position intermittent sequence photographs by illuminating the subject with a stroboscopic lamp.

All activities taking place within or passing through the plane of the camera slit are recorded in a position-time photograph where the ordinate is position and the abscissa, along the length of the film, is


Coupling to speed control knob at back of camera
Flex coupling to camera
Shock mount bracket
Camera drive centering adjustment



time. A precision of ½-mile per hour at 1000 miles per hour is possible. Unperforated 35-mm film is moved uniformly in either direction behind a vertical slit adjustable from zero to 0.270-inch wide in 0.001-inch increments. Film speed is infinitely controllable from zero to 100 inches per second with a Graham transmission, seen at the left of the view with the cover off, left, providing the variable film speed.

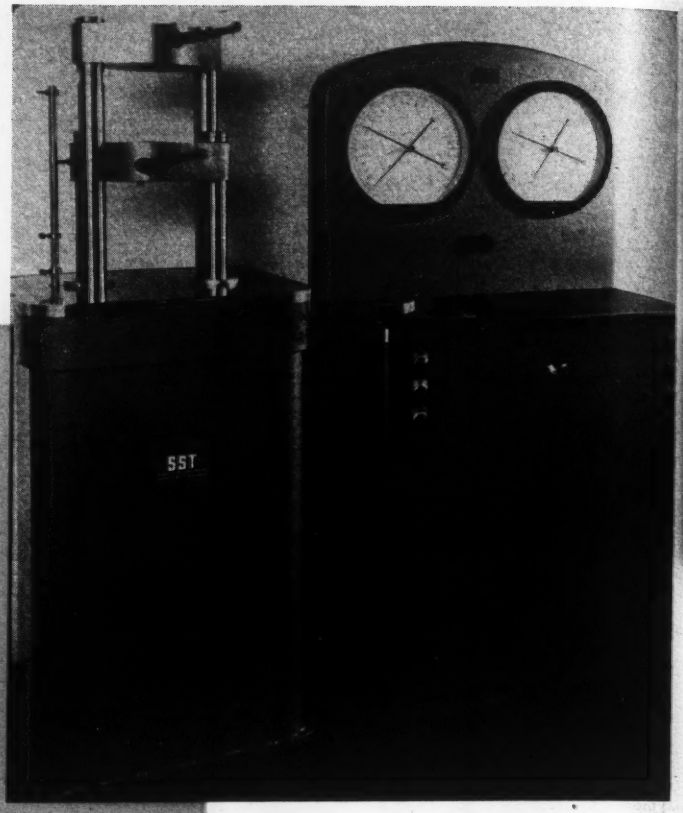
When in motion, the film receives three timing images in addition to the object image. A five digit number representing seconds and hundredths of seconds is supplied by a cumulative counter and short duration flash tube which is fired every 0.01-second and projected on the film as shown in the sketch, above. Pip marks projected on the film between these numbers indicate each thousandth of a second and can be interpolated to ten-thousandths. Auxiliary pips can be controlled by an external electric signal to give timing or reference marks as desired. Since the subject is photographed only during the time it is in the optical plane created in space by the projection of the slit through the camera lens, the resulting image on the film may resemble the object or may assume a variety of distortions, depending on relative velocities of film and image. Some examples of simple distortions are given in the sketches, left above, which show images produced on film moving at a fixed speed when the images are moving as indicated relative to film speed.

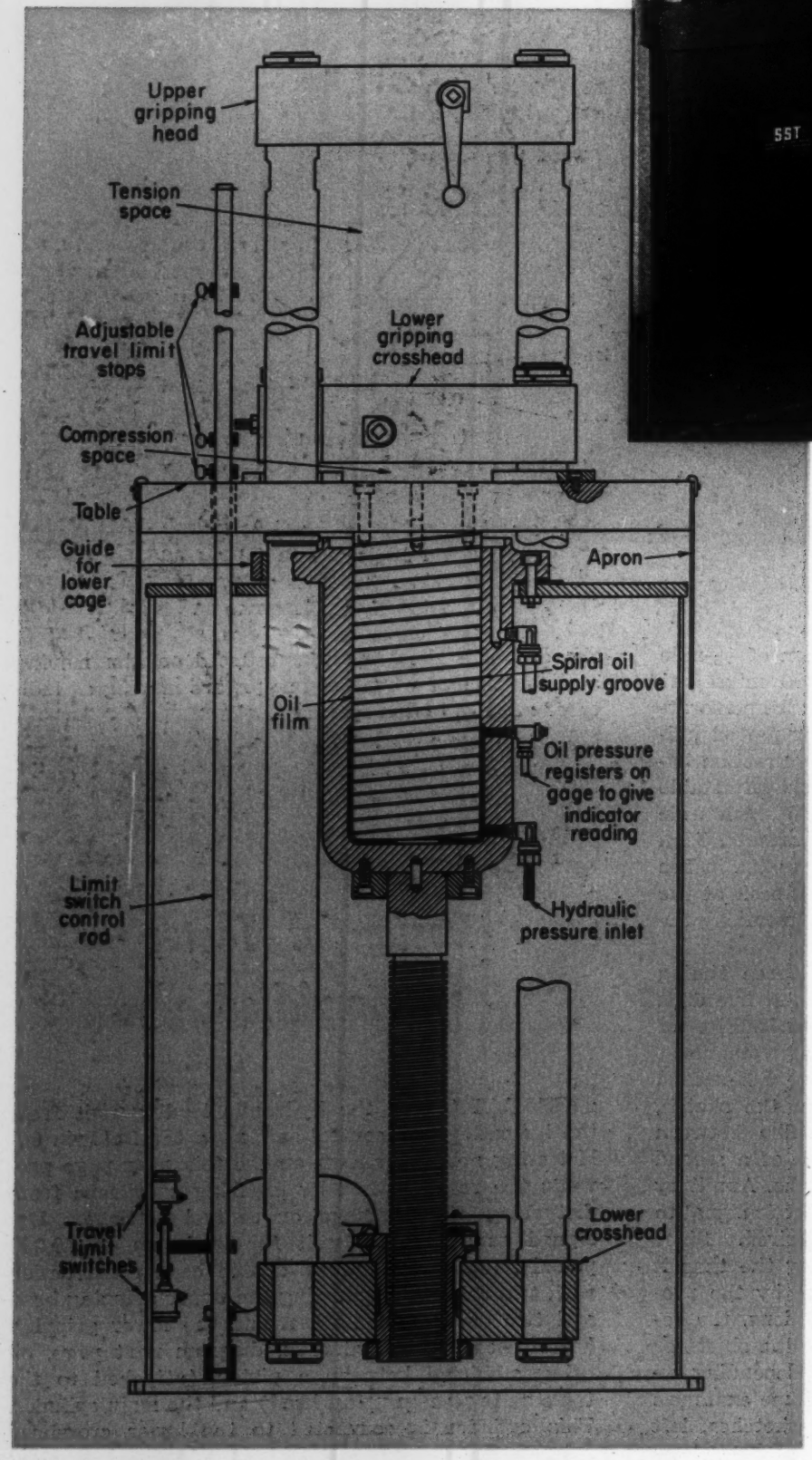
Design of Tester Simplified

SEPARATION of the hydraulic loading unit from the indicating and control section in the Baldwin 60-H testing machine shown on the following page prevents the recoil caused by breaking specimens from disturbing dial readings or control settings. Designed as a low-cost unit for loads up to 60,000 pounds, the tester uses a double cage construction with two columns supporting the upper gripping head and two other columns carrying the lower gripping head. Loading is hydraulic, diagram next page, oil being admitted between a piston (attached to the table and upper gripping head) and the main cylinder. This cylinder is connected to the lower crosshead

through a rapid traverse screw which is driven by a ¼-hp motor for prepositioning the lower gripping head a total of 17 inches at speeds to 10 inches per minute. The loading cylinder and piston are lapped, eliminating the need for seals; a spiral groove around the piston supplies lubrication between piston and cylinder, excess oil being returned to the reservoir.

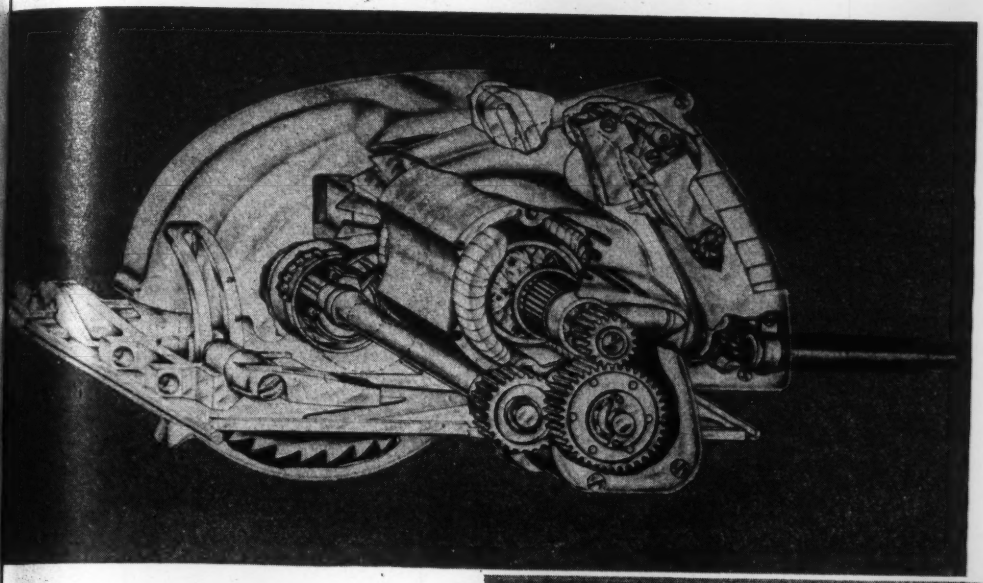
A Vickers directional speed control valve on the control cabinet provides for up or down head motion varying from 0 to 2 inches per minute; limit switches, shown in the drawing, protect the machine from accidental overtravel. Test loads are determined from two Bourdon gages (60,000 and 12,000-pound ranges) direct connected to the hydraulic pressure in the main loading cylinder. Gages are accurate within 1 per cent or 1 dial division, whichever is greater. Both hydraulic pump motor and the crosshead adjusting motor are controlled by start and stop pushbuttons on the indicator cabinet, with lights indicating when each motor is in operation.

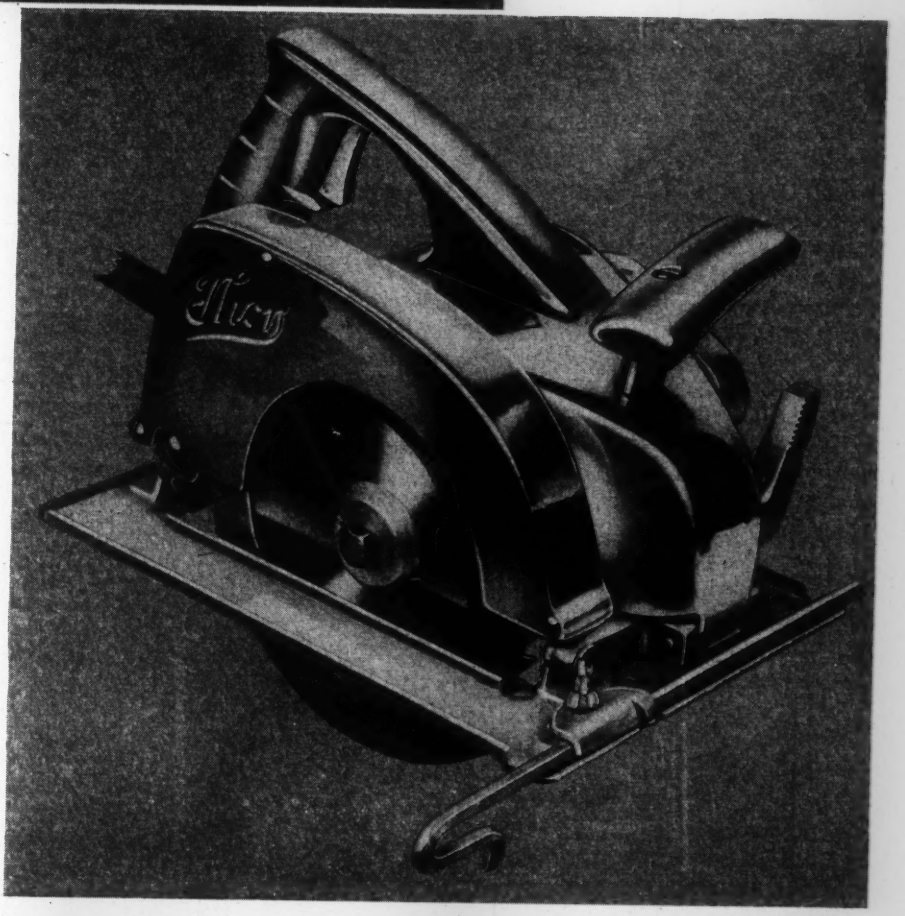




Geared Drive Powers Portable Saw

DIRECT gear drive from motor to blade combined with a long blade spindle having widely spaced ball bearings give the 6-inch portable electric saw shown in the il-

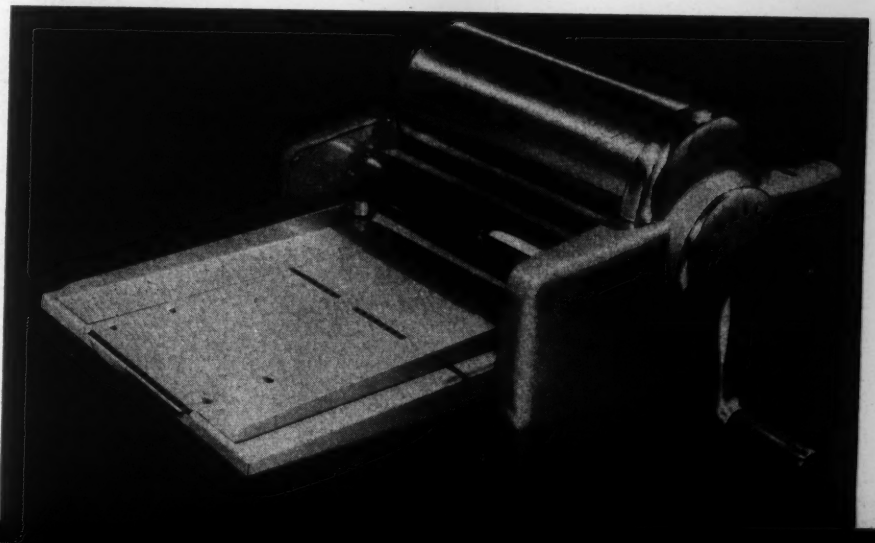




lustration, right, full power operation with no whip or wobble of the blade. Power loss through the double-reduction gearing shown in the cutaway drawing, above, is less than with other drive systems. Designed for safe operation and efficient servicing, the machine includes a safety-lock switch or lock pin on the handle that must be deliberately depressed before the trigger will operate, and an automatic spring-loaded blade guard. Brush holders are easily accessible and the blade can be changed in five seconds without removing guard or base. Depth and bevel scales retract into the housing when not in use. The saw, made by Independent Pneumatic Tool Co., has a 3600-rpm free speed, weighs 13½ pounds and makes a 2 1/32-inch deep, 90-degree cut or a 1⅜-inch, 45-degree bevel cut.

Duplicating Machine Redesigned

RESTYLING the case combined with a judicious selection of materials has resulted in improved appearance, in addition to substantial savings in manufacturing costs, of the Duplicopy Model H-44 machine shown at the right. Redesigned by Karl Brocken, the

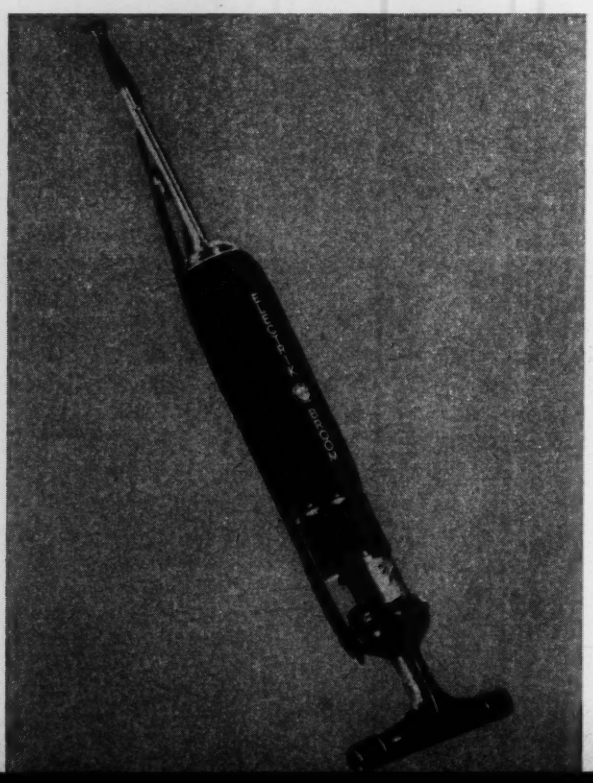


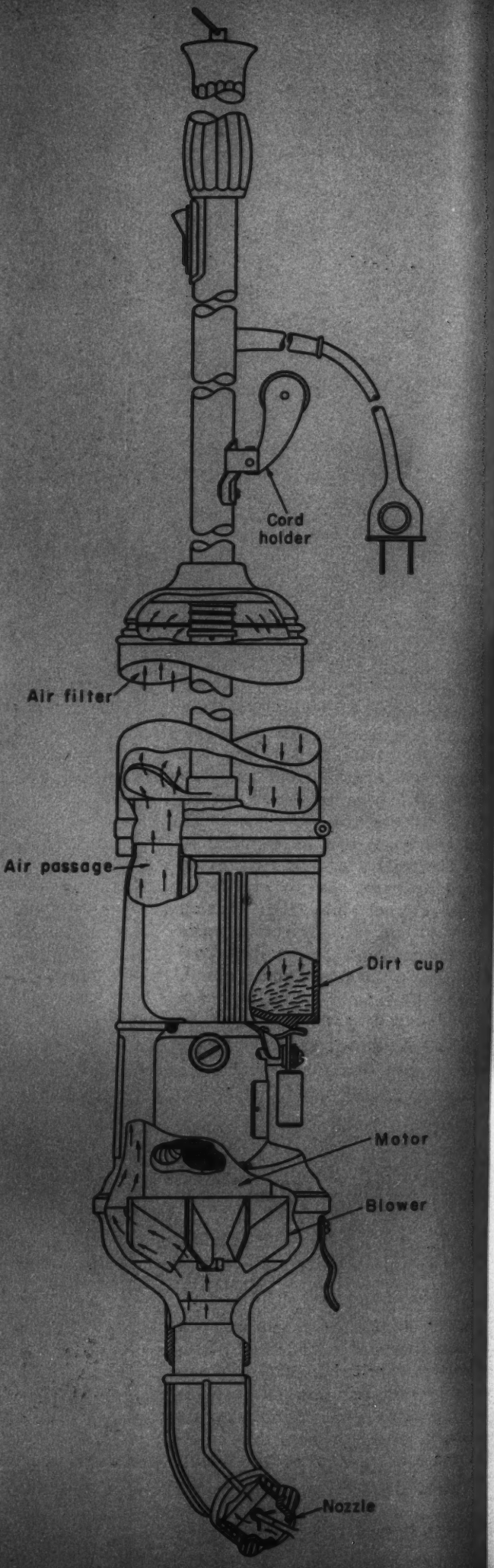
machine now utilizes low-cost diecast aluminum sides, handle, drum and internal parts, with trays and ink brackets of sheet aluminum.

The machine operates on the rotary principle with a floating pressure roller. As many as four colors may be reproduced at one time with the originals reusable at a later time. In the reproduction process, a sheet of carbon paper is placed in a reverse position behind the master sheet of 24-pound enamel paper. By typing, writing or drawing on the top side of the master sheet, a negative print is produced on the back of the sheet. This reverse master is clamped to the drum and, as the copy sheets are fed through the machine, the top surface of each is moistened with a special duplicating fluid which dissolves a small amount of carbon from the master onto the copy. Up to 300 copies can be made from one original.

Unique Vacuum Cleaner Design Cuts Weight

UNUSUAL design of the Electrikbroom home vacuum cleaner, shown at the right and below, combines blower, motor and dirt collector into an exceptionally compact, lightweight (6½ pound), one-piece unit. Made by the Regina Corp., the cleaner includes a snap-on removable plastic dirt cup located at the bottom of the air filter bag. Air flow is from the rubber covered nozzle, through the fan, past the motor and dirt cup through a tube or pipe, and into the filter bag where dirt falls into the removable cup.





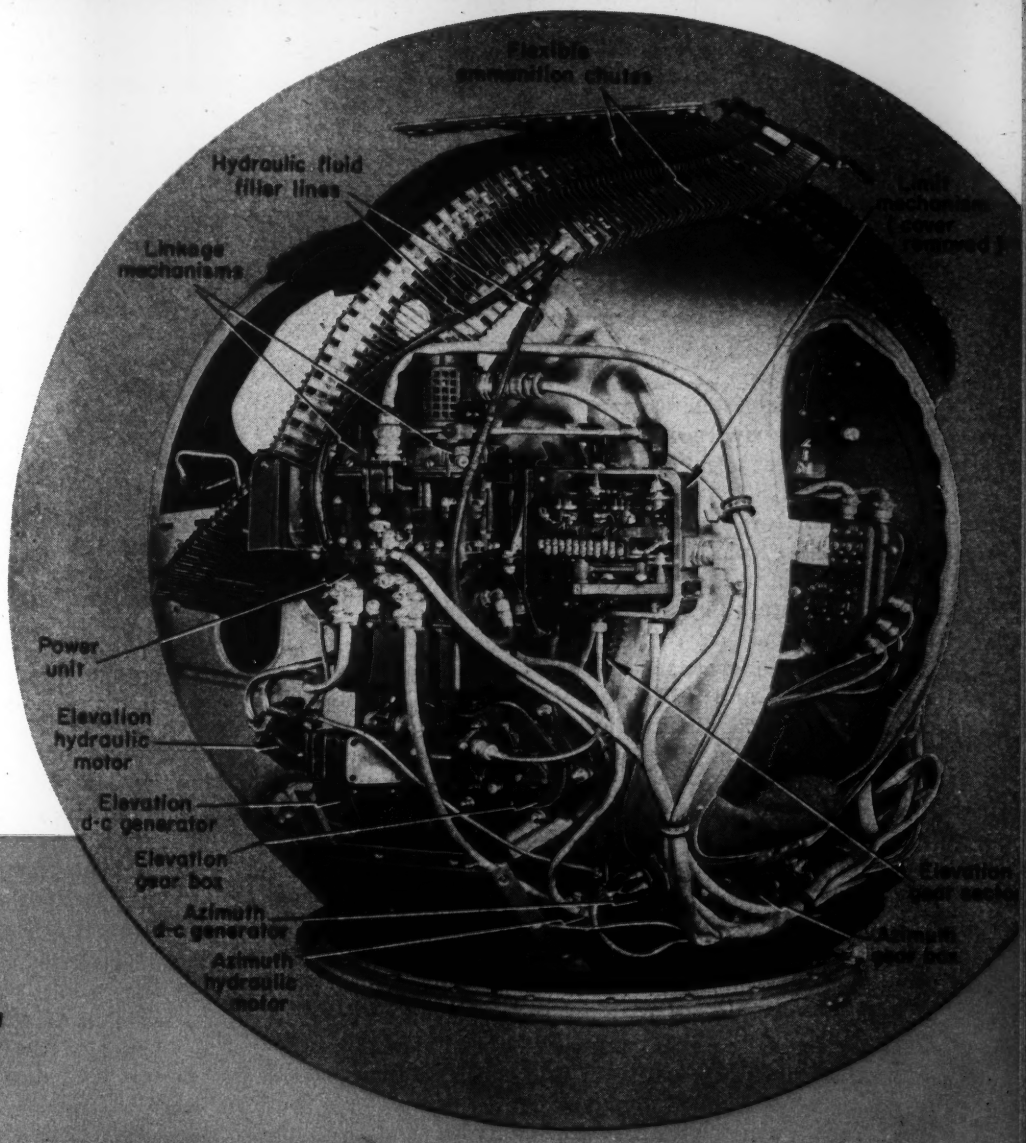


Fig. 1 — Application of hydraulic servomotors in a Sperry Gyroscope Co. aircraft turret. These drives represent an example of the problem of determining the proper motor size for a specific drive

By
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Cambridge, Mass.

What Size Motor?

. . . for proper operation of servomechanism

THE machine designer often is called upon to design equipment involving the use of servomechanisms or automatic control systems. For example, he may be asked to design a high-speed wire-drawing machine with provisions for maintaining constant tension on the take-up reels or an aircraft turret servo as shown in *Fig.* 1. He might also be called upon to design a contour milling machine for manufacturing parts of irregular shape under the control of a master pattern. In this case some provision would be necessary for automatically positioning various slides in accordance with the shape of the master pattern.

One major factor influencing his servo design is the size of motor required. It is obvious that horsepower alone is not a sufficient measure of capacity to drive a load such as is encountered in a servomechanism, since some of a motor's power is absorbed in accelerating itself. This article, therefore, will discuss the subject of motor-size determination under dynamic operating conditions.

The block diagram in *Fig.* 2 represents a positional servomechanism. Broadly speaking, the function of a positional servomechanism is to maintain input and output equal within prescribed limits in spite of load disturbances. It does this by measuring the error between input and output, the error forming a signal to actuate an amplifier-controller. In turn, the amplifier-controller operates the servomotor, or load-driving member, in a direction to diminish the error.

By appropriate design of the amplifier-controller, it is usually possible to make the over-all system stable and, servomotor permitting, to meet the over-all performance specifications.

The servomotor — a power-conversion unit that accepts energy from a local source and delivers energy to the servomechanisms's load — may be much the same as other motors, electric or hydraulic. Because it must be capable of control over extremely wide ranges of speed or torque, however, it is often more complex than an ordinary motor. As examples of servomotors, a hydraulic transmission and a Ward-Leonard drive are shown schematically in *Figs.* 3 and 4. These two types are commonly used in servomechanisms larger than one or two horsepower.

GEORGE C. NEWTON JR. acquired early experience in the development and design of servomechanisms as research assistant in the servomechanism laboratory of MIT and at the special Naval Ordnance plant in York, Pa. Later as a project engineer for Sperry Gyroscope Co., he worked primarily on development of aircraft fire-control equipment for bomber defense. In 1946, he returned to MIT where he is now principally engaged in teaching graduate courses in servomechanisms

The hydraulic transmission[1] utilizes a variable-displacement pump (driven at substantially constant speed) connected to a constant-displacement motor which drives the load. Speed of the motor is controlled by varying the displacement of the pump, the servomechanism's amplifier-controller being used to actuate the stroke (displacement) control of the pump. On the other hand, the Ward-Leonard drive consists of a shunt d-c generator (driven at substantially constant speed) connected to a shunt d-c motor which drives the load. In this case the speed of the motor is controlled by varying the generator's field current (which the servomechanism's amplifier-controller supplies). Frequently an Amplidyne or some other type of special generator is used in lieu of the simple shunt machine. Throughout this article the terms *hydraulic servomotor* and *electric servomotor* will refer to the hydraulic transmission and the Ward-Leonard drive, respectively.

Technical literature is replete with discussions of the design theory of servomechanisms.[2,3,4,5,6] Usually these discussions are based upon analysis in terms of linear differential equations and have the objective of finding the controller characteristics necessary to make a given servomotor and load perform satisfactorily. Modern servomechanism design theory is essentially compensation theory. Assuming a load and servomotor are given, it tells how to make amplifier-controllers work with these essentially "fixed" components. There is little discussion of the procedure to be used in selecting the servomotor upon which the remainder of the design is based.

This lack of emphasis on servomotor selection is probably the result of thinking that this part of design is obvious. Yet, because it is so fundamental to the whole problem of servomechanism design, a discussion of servomotor selection seems worthwhile. The size of the motor element is often an important factor influencing selection or rejection of a particular servomotor. For example, if a Ward-Leonard servomotor should turn out to be twenty-five times as heavy as a hydraulic transmission type of servomotor, the hydraulic transmission would be favored. How strongly the hydraulic transmission would be favored depends upon the importance of weight in the overall design.

CONDITIONS A MOTOR MUST SATISFY: A general solution to the problem of finding the conditions which a motor must satisfy to drive a given load would be complex. That this is so may be seen by considering the number of parameters that must enter into such conditions. The conditions under consideration will arise from the motor limitations caused by various kinds of saturation. In how many ways may a motor saturate? First, it is evident that a motor may saturate with respect to torque; it may be pos-

1. References are listed at end of article.

Table 1—Load Specifications and Parameters*

Load Description	Load Specifications: Peak Load Torque T_l (lb-in.)	Load Inertia J_l (in-lb sec²/rad)	Peak Load Velocity v_l (rad/sec)	Peak Load Accel. a_l (rad/sec²)	Load Parameters: Peak Load Torque $J_l a_l + T_l$ (lb-in.)	Peak Load Power P_l (in-lb/sec)	Load Time Constant τ_l (sec)
Ship's Rudder	1,800,000	80,000	0.080	0.040	1,803,000	144,000	2.0
Gun Turret	4000	800	2.0	20	20,000	40,000	0.10

* Numerical calculations use the inch-pound (force)-second system of units. Angular units are in radians. All equations and inequalities are correct in any consistent system of units; the particular system used here happens to be convenient for discussion of hydraulic machinery. For comparison purposes the more familiar power unit, the horsepower, is occasionally used; 1 hp is equivalent to 6600 in-lb/sec. With respect to moment of inertia, the value in pound-inches of torque per radian per second squared of acceleration is obtained from the value in inch-squared pounds by dividing the latter by the acceleration of gravity (386 in/sec²).

sible to get only a certain amount of torque out of a given motor regardless of how hard it is driven. Similarly, limitations may exist upon the maximum velocity which a motor can attain. It is possible for the velocity and torque saturations to be interrelated so that a functional curve governing these saturations would characterize the motor.

In addition to the velocity and torque saturations, there may exist saturation limits on the first n derivatives of the motor torque with respect to time. These may arise from saturation of other physical quantities in the motor or its drive such as, for example, a limited rate of change of field current in the generator of a Ward-Leonard drive. Interrelationships may exist among the various saturation limits of the torque derivatives.

For these reasons simplifying assumptions are necessary. In the solution to be presented, extreme simplification perhaps has curtailed the number of applications of the results that may be made legitimately. Nevertheless, by carrying through such a simplified solution, it will be possible to gain insight into the more general problem.

Three simplifying assumptions, used to establish the condition a motor must fulfill in driving a given load, are as follows:

1. An "ideal" gear train of over-all reduction ratio R is the coupling device used to connect the motor to its load. An ideal gear train is one having negligible inertia, no losses, no backlash, and negligible spring effect. At first sight it may seem that ignoring the inertia of the gear train is a serious error. The best justification for this assumption is to carry out the analysis based upon it, and then to check the errors committed in special cases. Such checks[10] frequently show that the error of ignoring the gear-train inertia is equivalent to an error of the order of two to five per cent in the value taken for the inertia of the load.

2. Four quantities are sufficient to specify the working load:
   a. Peak velocity, v_l
   b. Peak load acceleration, a_l
   c. Load inertia, J_l
   d. Peak load torque, T_l

Peak load torque is the sum of load torque and torque required to accelerate the load inertia.

It is supposed that the peak load torque, the peak load velocity, and the peak load acceleration all occur at random with respect to each other. Therefore, the extreme condition occurs when peak values are reached simultaneously.

3. Three quantities are sufficient to specify the motor:
   a. Peak motor torque, T_m
   b. Peak motor velocity, v_m
   c. Motor inertia, J_m.

Peak motor torque and the peak motor velocity are assumed to be independent of one another.

It is convenient to introduce four additional symbols. Let the peak motor power, P_m, be defined as the product of the peak motor velocity times the peak motor torque:

$$P_m = v_m T_m \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (1)$$

According to the assumptions, the peak motor power is the maximum power that the motor is capable of delivering. Next let the peak load power

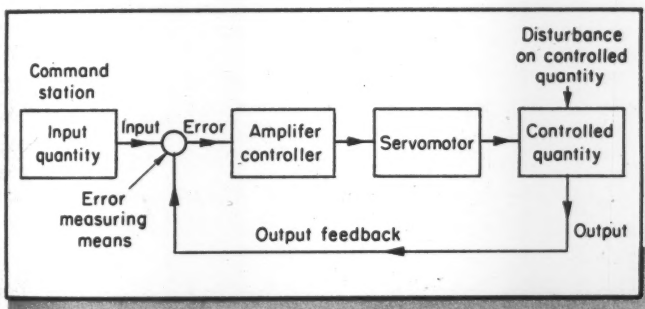


Fig. 2—Above—Block diagram of a servomechanism, with permission from Brown and Campbell (Reference 3)

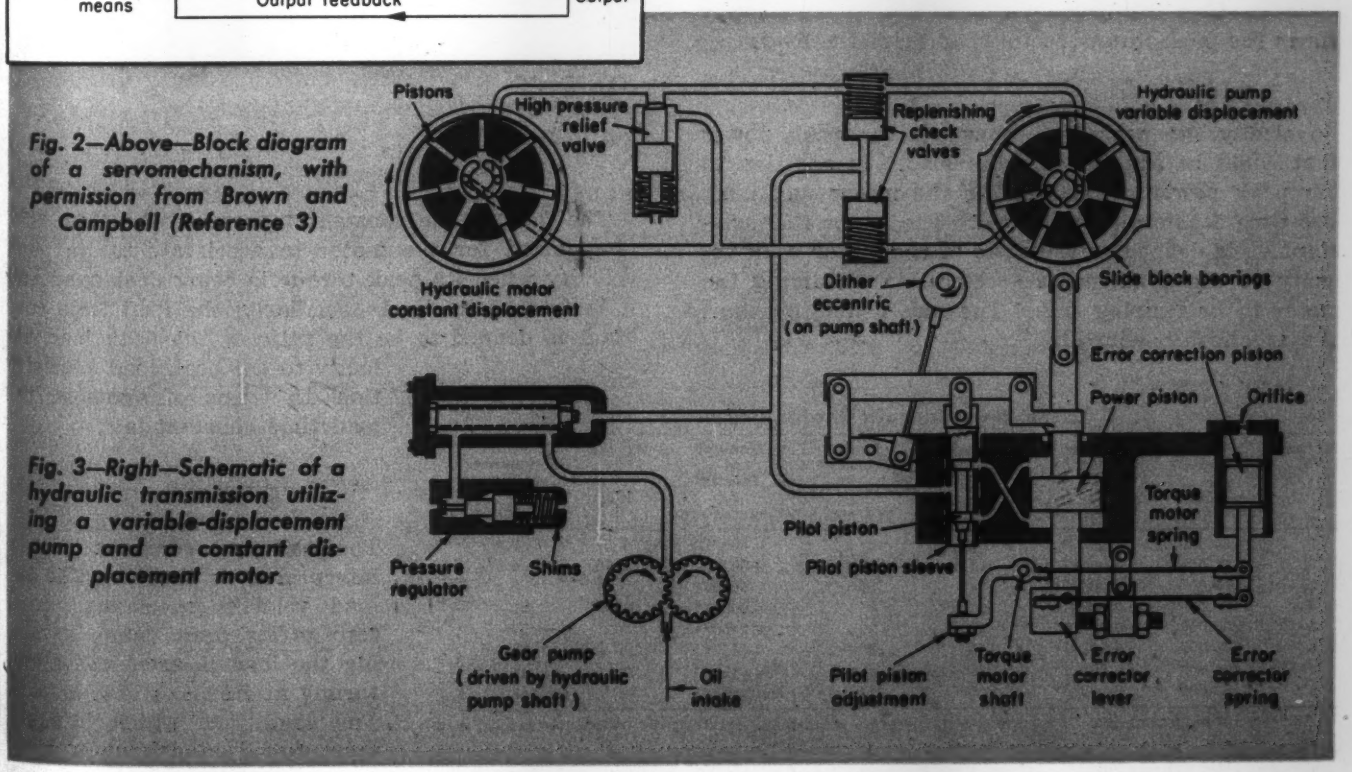


Fig. 3—Right—Schematic of a hydraulic transmission utilizing a variable-displacement pump and a constant displacement motor

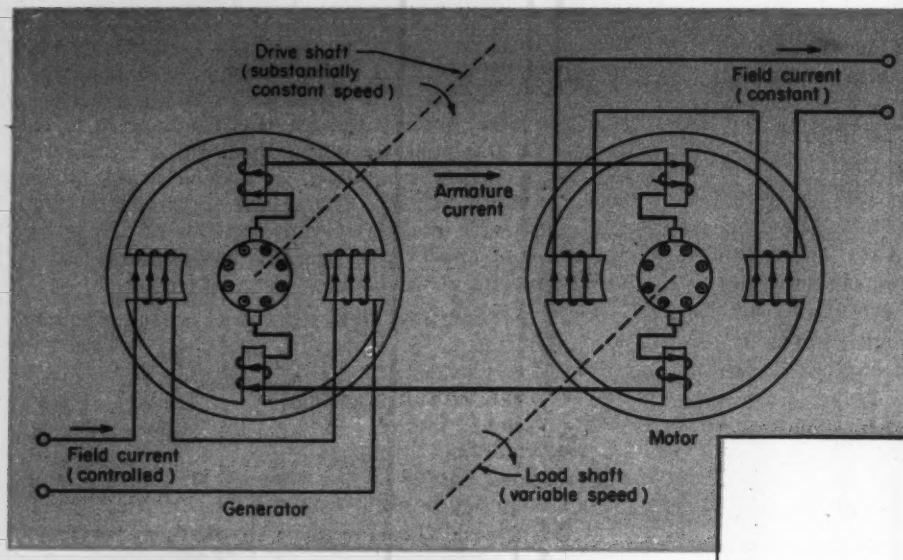
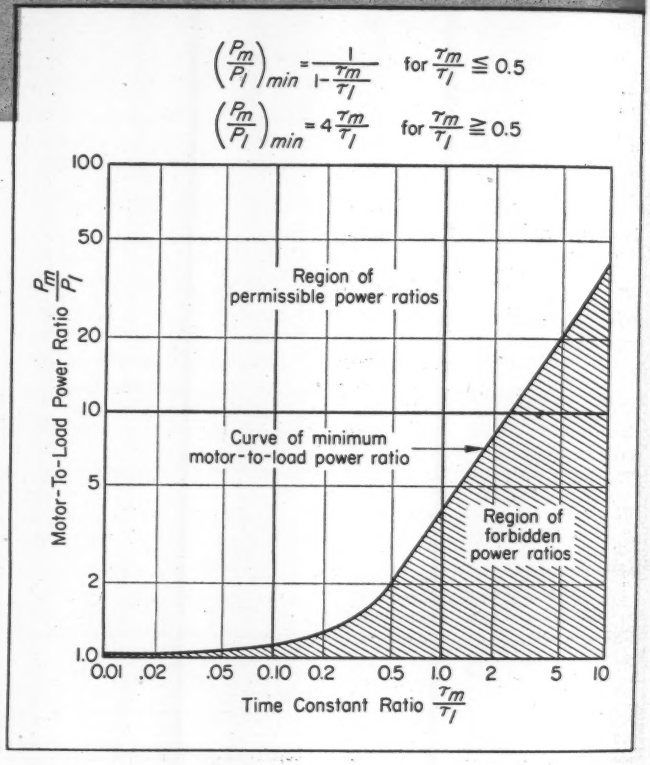


Fig. 4—Left—Schematic of a Ward-Leonard drive using a generator and a motor of the shunt direct-current types

Fig. 5—Below — Motor-to-load power ratio plotted against time-constant ratio, showing the region of permissible power ratios



Nomenclature

a_l = Peak load acceleration, rad/sec²
F_o = Overpower factor
J_l = Load inertia, in-lb sec²/rad
J_m = Motor inertia, in-lb sec²/rad
P_l = Peak load power, hp, or in-lb./sec
P_m = Peak motor power, hp or in-lb./sec
R = Gear reduction ratio of gear train connecting motor to load
T_l = Peak load torque, lb-in.
T_m = Peak motor torque, lb-in.
v_l = Peak load velocity, rad/sec
v_m = Peak motor velocity, rad/sec
W_m = Motor weight, lb
τ_l = Load time constant, sec
τ_m = Motor time constant, sec

be defined as the product of the peak load velocity times the peak torque required to drive the load:

$$P_l = v_l(J_l a_l + T_l) \quad \ldots\ldots (2)$$

Physically the peak load power is the peak power that could be absorbed by the load. In addition to the peak power definitions for the motor and load, two time constants are defined. The motor time constant τ_m is defined as the ratio of the motor peak velocity to the peak motor acceleration, assuming the motor to be running free. The motor time constant is symbolically defined by

$$\tau_m = \frac{v_m J_m}{T_m} \quad \ldots\ldots (3)$$

The motor time constant represents the time that is required for the motor to accelerate to its peak velocity when its peak torque is being developed and it is free of its load. Similarly, the load time constant is defined to be the ratio of the peak load velocity to the peak load acceleration. In terms of symbols, the load time constant is

$$\tau_l = \frac{v_l}{a_l} \quad \ldots\ldots (4)$$

The load time constant may be interpreted as the time for the load velocity to change from zero to its peak value, assuming the load is accelerated uniformly at its peak acceleration. The conditions which a motor must fulfill in driving a given

Table 2—Hydraulic Motor Data

Motor No.	Peak Power† P_m (in.-lb/sec)	Time Constant τ_m (sec)	Peak Torque† T_m (lb-in.)	Peak Velocity† V_m (rad/sec)	Moment of Inertia J_m (in.-lb sec²/rad)	Weight W_m (lb)	Nominal Horsepower (Continuous) (hp)
1	33,800	0.00280	90	376	0.000670	2.5	3.4
2	47,500	0.00407	121	392	0.00126	3.75	4.8
3	53,800	0.00382*	143	376	0.00145*	4.0	5.4
4	172,000	0.00257*	456	376	0.00311*	20	17
5	265,000	0.00488*	903	293	0.0150*	30	27
6	265,000	0.00299*	1155	230	0.0150*	30	27
7	476,000	0.00617*	2285	209	0.0673*	75	48
8	480,000	0.00396*	2860	168	0.0673*	75	48

† For an industrial line of constant-displacement, axial-piston units, peak ratings based on a 5 per cent duty cycle.
* Based on estimates.

Table 3—Electric Motor Data* (Estimated)

Motor No.	Peak Power P_m (in.-lb/sec)	Time Constant τ_m (sec)	Peak Torque T_m (lb-in.)	Peak Velocity V_m (rad/sec)	Moment of Inertia J_m (in.-lb sec²/rad)	Weight W_m (lb)	Nominal Horsepower (Continuous) (hp)
1	33,000	0.0617	158	209	0.0466	60	1
2	66,000	0.0738	316	209	0.112	100	2
3	165,000	0.0948	790	209	0.358	200	5
4	330,000	0.083	2100	156.5	1.12	400	10
5	660,000	0.108	4200	156.5	2.90	750	20
6	1,650,000	0.120	10,500	156.5	8.04	1500	50

* For an industrial line of d-c shunt units with interpoles but without poleface windings. Peak ratings based on a 4 per cent duty cycle.

load are now set forth. These several conditions have been derived on the basis of the assumptions stated and shown in the appendix at the end of this article. The conditions simply state that the ratio of the peak power of the motor to the peak power of the load must be greater than or equal to a certain function of the ratio of the time constant of the motor to the time constant of the load. Analytically, the conditions are expressed as

$$\left(\frac{P_m}{P_l}\right)_{min} = \frac{1}{1-\frac{\tau_m}{\tau_l}}; \text{ when } \frac{\tau_m}{\tau_l} \leqq \frac{1}{2} \quad \ldots\ldots (5a)$$

$$\left(\frac{P_m}{P_l}\right)_{min} = 4\frac{\tau_m}{\tau_l}; \text{ when } \frac{\tau_m}{\tau_l} \geqq \frac{1}{2} \quad \ldots\ldots\ldots (5b)$$

Torque to inertia or torque squared to inertia ratios for the motor do not appear in this result. For a condition involving the torque squared to inertia ratio of the motor, the reader is referred to Harris's paper[7] on "A Comparison of Two Basic Types of Servomechanisms."

A plot of the minimum power ratio is shown in *Fig.* 5. This plot shows that, for very large ratios of peak load velocity to peak load acceleration relative to the ratios of peak motor velocity to peak motor acceleration (that is, for small values of the time-constant ratio), the peak motor power will be of the same order of magnitude as the peak load power. Under these conditions, it is evident that merely matching the motor power to the load power requirement is a satisfactory way of solving for motor size. As soon as the time-constant ratio becomes appreciable, however, the motor must be provided with a peak power capacity which is greater than the power rating of the load.

It should be noted that the area above the curve of *Fig.* 4 represents a region of permissible power ratios, whereas the area below the curve represents an area of forbidden ratios. If, for a given time-constant ratio, the motor peak power to the load peak power falls below the curve of *Fig.* 5 into the region of impossible power ratios, then it is certain that the design will fail to meet one or more of the load specifications. In other words, the load will fail to obtain the peak velocity specified or the peak acceleration specified or both.

The method used to obtain Relations 5*a* and 5*b*, the basic result of this article, is as follows: On the basis of the assumptions stated, two inequalities involving the gear ratio between the motor and load may be set up. One inequality expresses the fact that the peak motor torque less the peak torque required to accelerate the motor inertia must be greater than or equal to the peak torque required to drive the load as reflected through the gear train. The other inequality expresses the fact that the motor peak velocity must be greater than or equal to the load peak velocity as reflected through the gear train. For a given motor and load combination, these fundamental inequalities represent constraint on the choice of gear ratio. In order for a motor to drive a load, it is necessary that the fundamental inequalities permit at least one gear ratio to exist and that this gear ratio be a real number. Making use of this fact allows the basic inequality to be derived by algebraic manipulation from the funda-

Table 4—Motor-Size Determinations

Application	Peak Load Power P_l (in.-lb/sec)	Load Time Constant τ_l (sec)	Motor No.	Peak Motor Power P_m (in.-lb/sec)	Motor Time Constant τ_m (sec)	Motor-to-Load Time Constant Ratio (τ_m/τ_l)	Minimum Motor-to-Load Peak Power Ratio $(P_m/P_l)_{min}$	Minimum Allowable Motor Peak Power $(P_m)_{min}$ (in.-lb/sec)	Over-Power Factor F_o
Turret with hydraulic drive .	40,000	0.10	TABLE 2 No. 2	47,500	0.00407	0.0407	1.041	41,600	1.15
Turret with electric drive (First trial)...	40,000	0.10	TABLE 3 No. 2	66,000	0.0778	0.778	3.11	124,000	0.53
Turret with electric drive (Second trial)..	40,000	0.10	TABLE 3 No. 3	165,000	0.0948	0.948	3.79	152,000	1.08
Rudder with hydraulic drive .	144,000	2.0	TABLE 2 No. 4	172,000	0.00257	0.00128	1.001	144,000	1.19
Rudder with electric drive ..	144,000	2.0	TABLE 3 No. 3	165,000	0.0948	0.0474	1.047	151,000	1.09

mental torque and velocity inequalities.

MOTOR-SIZE DETERMINATION: To determine motor size, a trial and error procedure may be used:

1. The peak power of the load P_l and the load time constant τ_l are computed by Equations 2 and 4 from the application specifications.
2. A preliminary figure for peak motor power P_m is estimated. This may be taken to be the peak load power multiplied by a factor equal to one plus a fraction indicated by the designer's experience.
3. Referring to data tables for the line of motors under consideration, the motor with the least greater peak power than the estimated figure is tentatively chosen.
4. The ratio of the motor time constant to the load time constant for this motor is computed.
5. By the use of *Fig.* 4 or Relations 5*a* and 5*b*, the minimum motor-to-load peak-power ratio for this time-constant ratio is found.
6. Using this minimum-power ratio, the minimum allowable peak motor power is computed.
7. If this minimum peak power is not less than the product of the peak power of motor tentatively chosen times a suitable "overpower" factor F_o, the procedure is repeated using a larger motor.

To illustrate the procedure in the preceding paragraph, numerical solutions are carried through for the two sets of load specifications given in TABLE 1. These specifications are representative of the rather severe requirements that might be met in aircraft gun-turret work and the normal requirements that might be found for a 10,000-ton merchant-ship rudder. In both cases the peak acceleration specification was set by the special operating requirement of limit-stop operation. It is not unusual for motor size to be critically affected by a load specification arising from such special operating conditions. On the basis of these load specifications, the peak load power and load time constant are determined by Equations 2 and 4 and listed in TABLE 1.

It will be assumed that the object of the numerical investigation is to compare the sizes of hydraulic and electric motors required to drive these loads. In order to do this, representative data for the lines of motors in question must be available. It will be assumed that the required data for a line of industrial, constant-displacement, axial-piston, hydraulic motors and a line of industrial d-c electric motors are shown in TABLES 2 and 3. The hydraulic motor data are for the Vickers Inc. line based on their published data except where such data were lacking. On moments of inertia, in particular, it was necessary to estimate frequently as indicated by the asterisks in TABLE 2.

The effect of fluid inertia is similar to the inertia of the rotating parts of a hydraulic motor and may be accounted for to a first approximation by increasing the inertia figure used for the rotating parts. Obviously, the fluid-inertia effect depends upon the geometry of the lines delivering oil to the hydraulic motor. For a particular transmission[1] where the fluid inertia was larger than in most servomechanism applications, the correction for fluid inertia was only about 5 per cent. Because this correction is generally small and because it depends upon the geometry of parts external to the motor, fluid-inertia correction factors are not included in the inertia figures of TABLE 1.

The electric motor data have been estimated on the basis of design data given by Kloeffler, Brenneman, and Kerchner[8], and by Gray[9]. This was done because of the lack of sufficient information on any one manufacturer's line of motors. Measurements made on a few motors in the lower power range tend to confirm the trends indicated by the data listed in TABLE 3. Because of the number of estimates involved, all the data presented for hydraulic and electric motors should be regarded as indicating order of magnitudes and should not be accepted as precise information upon which to base final designs. They should be satisfactory, however, for preliminary design studies.

Table 5—Results of Motor-Size Determinations*

Application	Peak Load Power	Minimum Tabulated-Size Hydraulic Motor		Minimum Tabulated-Size Electric Motor	
		Peak Motor Power	Motor Weight	Peak Motor Power	Motor Weight
	P_l (hp)	P_m (hp)	W_m (lb)	P_m (hp)	W_m (lb)
Turret	6.06	7.20	3.75	25.0	200
Rudder	21.80	26.60	20.00	25.0	200

* Peak power ratings are given in horsepower in this table to facilitate comparison. Power values in horsepower are obtained by dividing values in inch-pounds per second by 6,600.

Specific Ratings Obtainable from Supplier

Data upon which to base final designs should be obtained from the potential suppliers of the motors selected. Caution will have to be exercised in arriving at peak ratings. Until standards are developed for establishing the peak ratings of motors used in servomechanism applications, it will be the responsibility of each servomechanism designer to determine just how he will rate a specific piece of apparatus in view of his particular application.

Using the load parameters listed to the extreme right in TABLE 1 in conjunction with the motor parameters listed in TABLES 2 and 3 enables TABLE 4 to be constructed for the determination of both the electric and hydraulic motor sizes required in each of the two applications. For purposes of these examples, a motor size is considered correct if it is the smallest size listed in TABLE 2 or 3 that gives an overpower factor greater than unity regardless of how little the margin, even though a minimum overpower factor may be specified as part of the design requirement. Some margin is required to allow for a gear ratio tolerance[10].

Results of the motor-size determinations in TABLE 4 are summarized in TABLE 5 where peak power ratings (in horsepower) and weights are listed for the minimum available sizes of motors able to drive the loads. The peak rating of 25 horsepower, corresponding to a continuous rating of the order of 5 horsepower, for the electric motor used in the rudder application is lower by a factor of three or more than the ratings actually used. This is for two reasons. First, practically no margin of overpower is provided in the design. Second, no coupling losses are accounted for; most electric rudder drives use worm-

(*Continued on Page* 200)

Centrifugal Casting for Critical Components

By Herbert Cooper
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Fig. 1—Tapered jet engine ring cast centrifugally from stainless steel, rough cast, rough machined and finished

RECENT developments in the techniques for casting stainless steel centrifugally in permanent metal molds have opened new avenues for design engineers concerned with the creation of units that are cylindrical in shape. Although these methods were originally developed as a means for meeting the rigid requirements of jet aircraft engines, *Fig.* 1, the process holds promise for many other machine applications such as pump liners, seat rings and other critical cylindrical components.

Fundamentally, centrifugal casting can be divided into three broad categories—centrifuging, centrifugal pouring and true centrifugal castings. The first of these, centrifuging, is generally restricted to castings of small and rather complex shape as it is accomplished by grouping several castings around a central gate and rotating the entire mold. Centrifugal pouring is generally used in producing castings of symmetrical shape relative to the axis of rotation, and is particularly well adapted to the production of castings with thin metal walls. Thousands of aircraft turbosupercharger components have been produced by this method to assure top-quality castings with a remarkably low rejection rate.

True centrifugal casting involves not only the feeding of the casting by centrifugal force but also the shaping of it. Although pipe and bushings have been cast in this way in sand for many years, the first important use of this method with metal molds was stimulated by requirements for jet engine rings.

From a metallurgical standpoint, the advantages of centrifugal casting have long been known. Tests

have shown that the centrifugal action tends to drive the heavy metal outward, while the lighter inclusions such as slag, dirt and gas move inward. This separation of the inclusions from the metal results in denser and sounder castings. It is also known that the rapid chilling induced by solidification of the molten metal against a heavy metal wall produces a finer grained metal structure with improved physical properties. *Figs.* 2 and 3 show the finer grain achieved through this method as compared with that achieved through static casting. The following, quoted from a laboratory research report, points out the advantages offered by this method.

"A comparison of the two methods of casting type 18-8S stainless steel shows that centrifugal castings have better ductility with slightly lower strength both in the as-cast and water-quenched condition. The macrostructure, *Fig.* 2, shows that the metal cast centrifugally has smaller grains which are more uniformly distributed. The microstructure, *Fig.* 3, reveals finer, more well distributed pools of ferrite, a condition which should result in a lesser tendency toward intergranular corrosion, although of course, this feature would be absent in completely austenitic type alloys. In this latter case, however, the finer grain size of the centrifugally cast metal would be of benefit in reducing intergranular corrosion. Centrifugally casting this alloy would, in addition to the above, result in a denser metal, greater freedom from dirt and the absence of center line shrinkage with its resultant weakness."

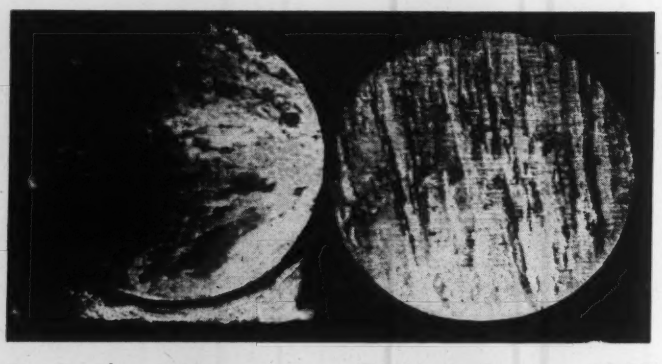


Fig. 2—Above—Macrostructure of cast stainless type 18-8S by static method, left, and centrifugal method, right

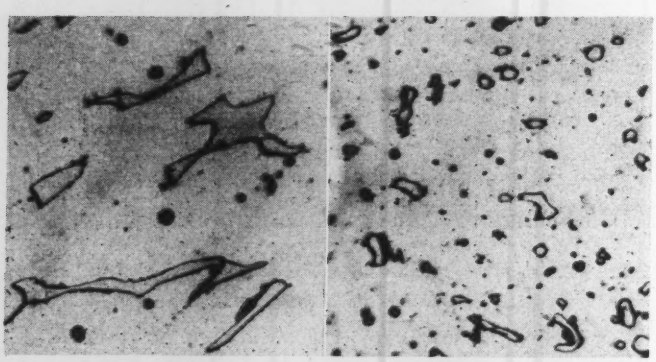
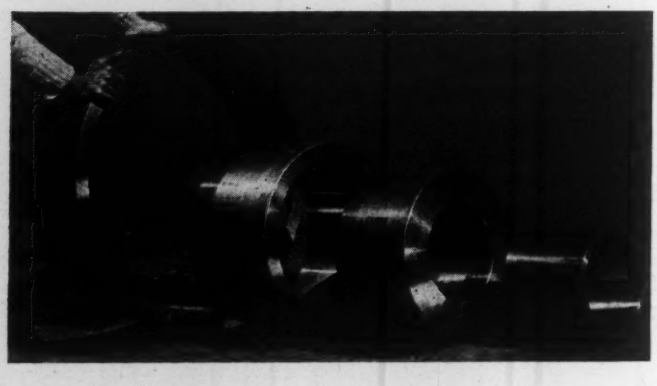


Fig. 3—Above—Microstructure of cast stainless type 18-8S showing static-cast structure at left and centrifugal at right

Fig. 4—Below—An assortment of centrifugal castings for Pratt & Whitney aircraft jet engine ring components



The Cooper Alloy Foundry Co. has been pouring stainless steel castings centrifugally in permanent metal molds since December, 1947. It is anticipated that this process will play an increasingly important role in American industry because it is ideally suited to the production of top-quality cylindrical shapes. Basic design considerations which indicate that this process will continue to grow in importance are outlined in the following paragraphs.

Factors in Design

High Casting Quality: The use of steel molds eliminates sand and dirt inclusions. The centrifugal force of 50*g* eliminates gas. Remarkably low rejection rates are maintained even where specifications require 100 per cent x-ray inspection with no welding permitted.

Increased Ductility and Exceptionally Fine Grain Structure: The rapid solidification provided for by the metal mold results in excellent physical properties.

Lower Unit Costs: Lower unit costs are possible with this method because the rejection rate is so remarkably low, thereby substantially increasing the yield from each heat. In addition cleaning costs are cut to a minimum due to the excellent surface condition which results.

Applications: This process is ideal for cylindrical rings which are obtained in quantity by slicing a centrifugally cast cylinder. It has also proved economical for the casting of bushings which are either straight or flanged on one end. It is ideal for tubular shapes.

Design considerations: Cost of the metal molds is relatively high and to assure economy, quantity production is a must. By the vertical-axis method high-alloy centrifugal castings with outside diameters ranging from 8 to 40 inches are being produced, with the upper limit constantly being raised. Current production includes rings with wall thicknesses varying from ½ to 4 inches and maximum height is considered to be about 15 inches.

A 180-HP FLUID magnetic clutch, believed to be the largest yet constructed, will be built by Duncan & Bayley Inc. for the NACA. The 5100 ft-lb torque device will be used as a control means for model studies in a new supersonic wind tunnel project.

PRODUCTION AND DESIGN
Modern Practices in Manufacture

Redesign for More Economical Manufacture

By James Williams
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The Warner & Swasey Co.
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THE best time to redesign for production is while a part is being designed. Many useful suggestions may come from the people who tool up jobs or who process them through the plant. A slight design change may allow a part to be made by an entirely different and much faster method.

Need for redesign on a large scale seldom should occur and action taken without sufficient consideration may not be beneficial. Such situations can be an indication of a bad spot in an organization—people not pulling together, lack of ability, etc.

There is no given path for anyone to follow in seeking improved production design. Merely making a design better than the old one is not the best measure. It is preferable to compare a design with what it should be and should cost. Actual examples of parts redesigned for both better performance and lower cost are discussed in the following paragraphs with specific details regarding their various functions and design features.

A machine housing that previously had been made of sheet metal with cast-aluminum doors is shown in

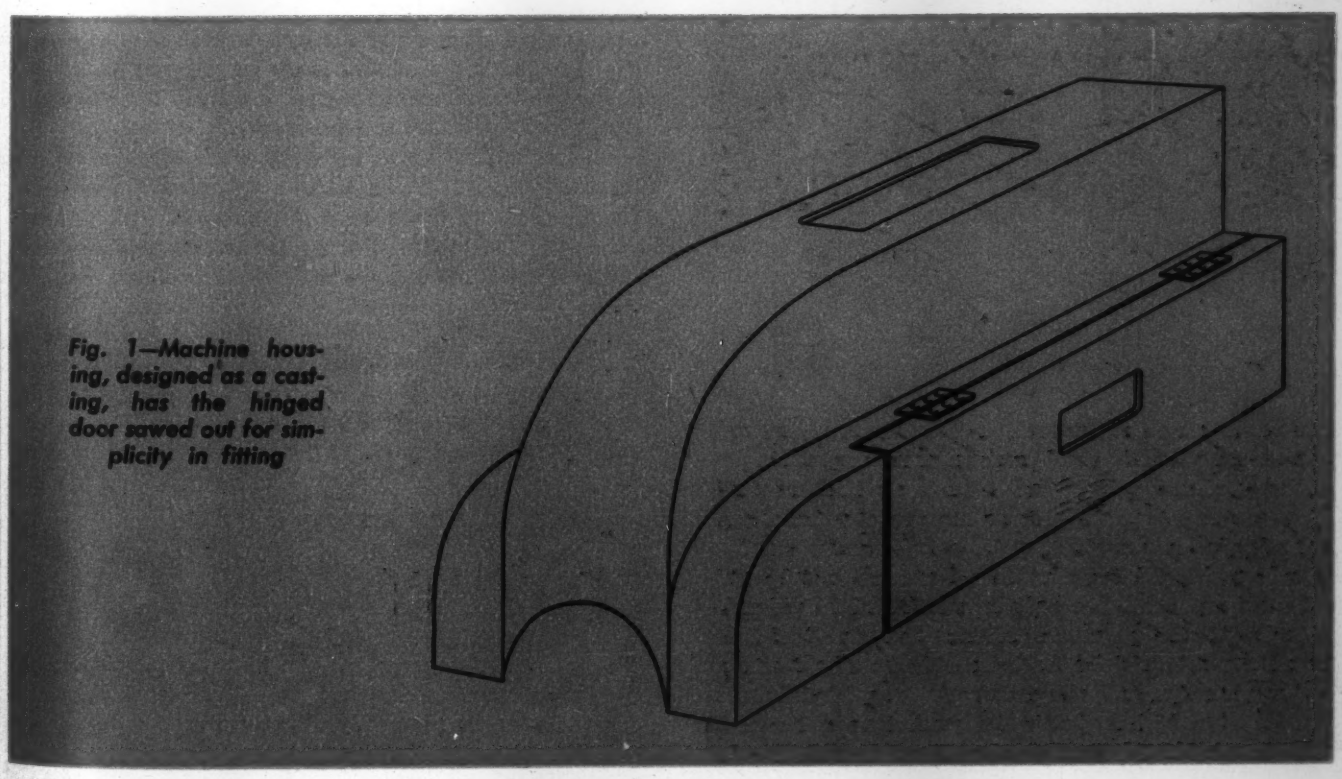


Fig. 1—Machine housing, designed as a casting, has the hinged door sawed out for simplicity in fitting

Fig. 1. Getting the sheet-metal housing to fit the machine and hinging and aligning the aluminum doors was a tedious job. Casting the entire cover of aluminum and sawing out the doors saved considerable time and the doors always matched. The one-piece casting turned out to be better looking, cheaper and more durable. (Twenty hours were saved in matching and fitting.)

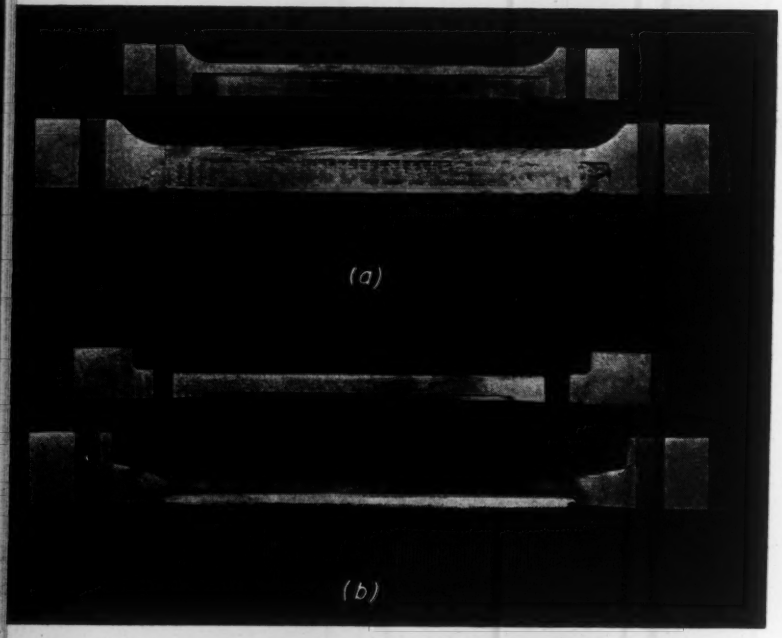


Fig. 2—Above—Faller bar comb for a textile machine redesigned from machined tool steel as at (a) to flash welded assembly (b) reduced materials cost as well as labor required

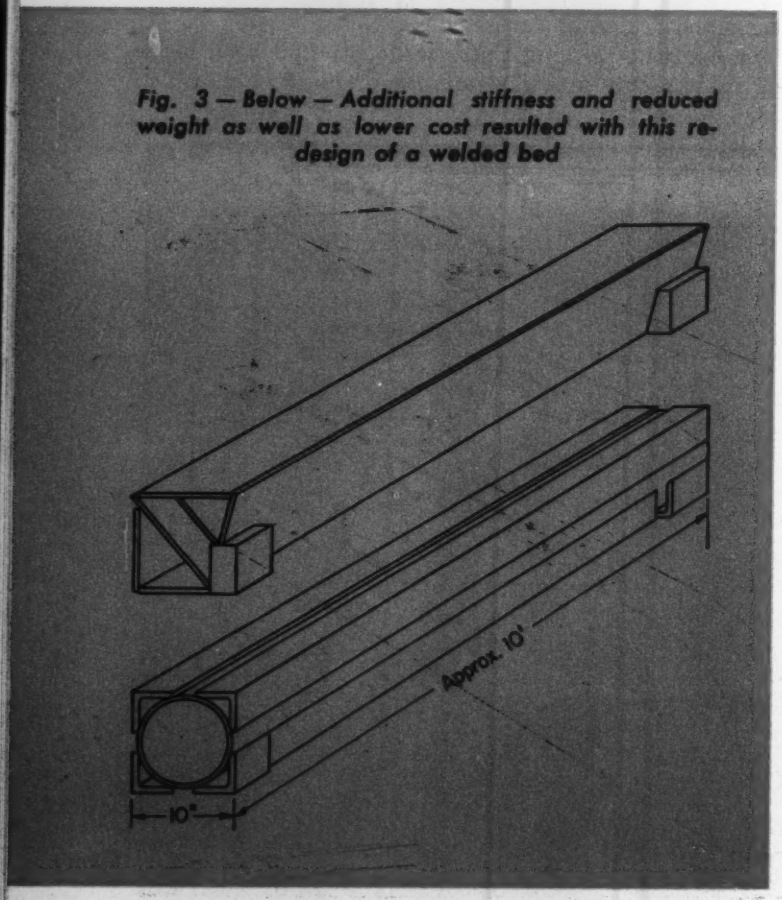


Fig. 3 — Below — Additional stiffness and reduced weight as well as lower cost resulted with this redesign of a welded bed

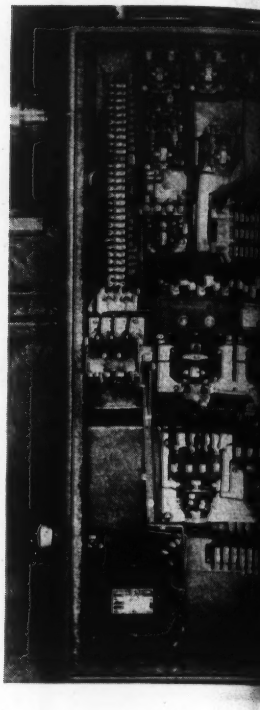


Fig. 4 — Control panel (a) widened as at (b) reduced wiring time by 25 per cent

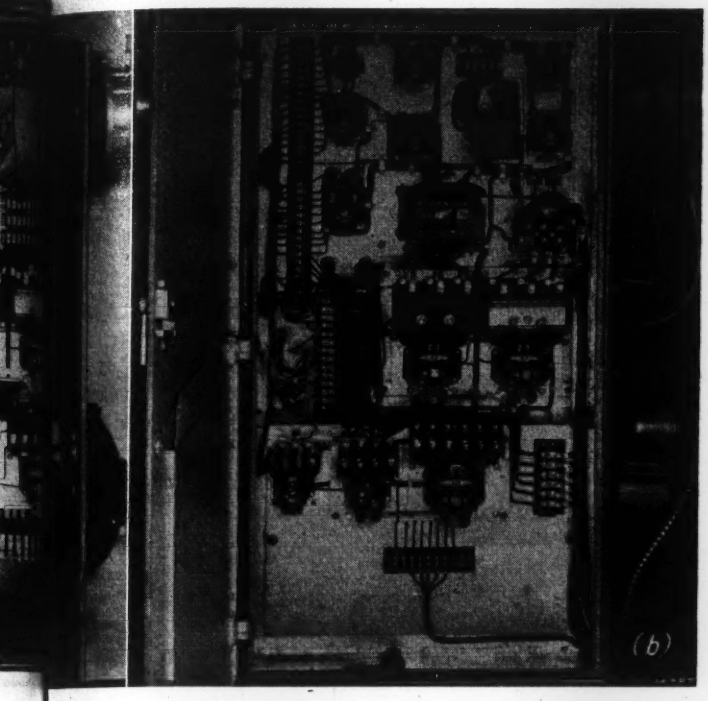
Fig. 2 is a faller bar or comb used on a textile machine. The original bar was machined from a solid piece of tool steel with the needles soldered against the face of the bar. The redesigned bar has a folded sheet-metal center section and tool steel ends. The three pieces are flash welded together and the needles are anchored in the slot with solder. The savings on this particular part were the large reduction in the amount of tool steel used and the time required to whittle the part from the solid. Hardening the parts by induction heating instead of a lead pot proves four times as fast.

A fabricated bed for a textile machine is shown in *Fig.* 3. The original was made of sheared plates, the stiffeners and blocks being hand welded and the longitudinal edges done on an automatic welder. The redesigned bed is made up with a standard tube and standard angles stitch welded by hand. The material costs for the two were about equal, with the savings being in fabricating time. Other advantages of the new over the old are additional stiffness and less weight, both fundamental design requirements for the part.

It was possible to close the ends on the new bed by welding on steel castings that had clearance holes cast in place, saving two boring mill operations on a very cumbersome part.

Size Increase Cuts Cost

An original control panel, *Fig.* 4*a*, used on a small electric turret lathe is shown redesigned in *Fig.* 4*b* with the only change being an increase in width of the box. An increase of just 3½ inches reduced the assembly time on the box by 25 per cent by giving adequate room to work. This type of redesign has the wholehearted support of the man doing the as-



sembly job as well as that of the field engineers and servicemen.

A mechanism designed to change horizontal motion to vertical, *Fig. 5a,* consisting of two racks and a gear segment as shown was expensive in both time and tolerances. A new mechanism for performing the same functions, *Fig. 5b,* employs a stamped bell-crank and a small link riveted to the vertical and horizontal members. Savings in machining time, fitting and inspection on the new design are tremendous. The parts are much lighter and easier to move, resulting in less power consumption.

Functional Alternatives Necessary

Original design of a cam and roller assembly, *Fig. 6a,* worked very well but the cost of manufacture was prohibitive. The levers and cams were forgings requiring considerable machining, particularly the face-cam tracks. Holding close tolerances and avoiding warpage of the track after heat treating were expensive. On the redesigned mechanism for doing the same job, the levers as well as the cams are stampings. Instead of face cams to control all movement of the levers, a cam and counter cam achieves the same effect with considerable weight reduction. The flat cams, which are stampings, can be stacked on an arbor and cut several at a time on a cam lathe, cam miller or cam shaper. The total amount of work involved is only a small fraction of that required for the original and the arrangement functions equally well.

In these actual cases, redesign has meant real dollars-and-cents savings and the predominate change invariably was in the direction of simplicity. Simplicity in design is the result of good basic design thinking. It is hoped that perhaps some of these examples will start some thinking along the same lines in other areas of machine design activity.

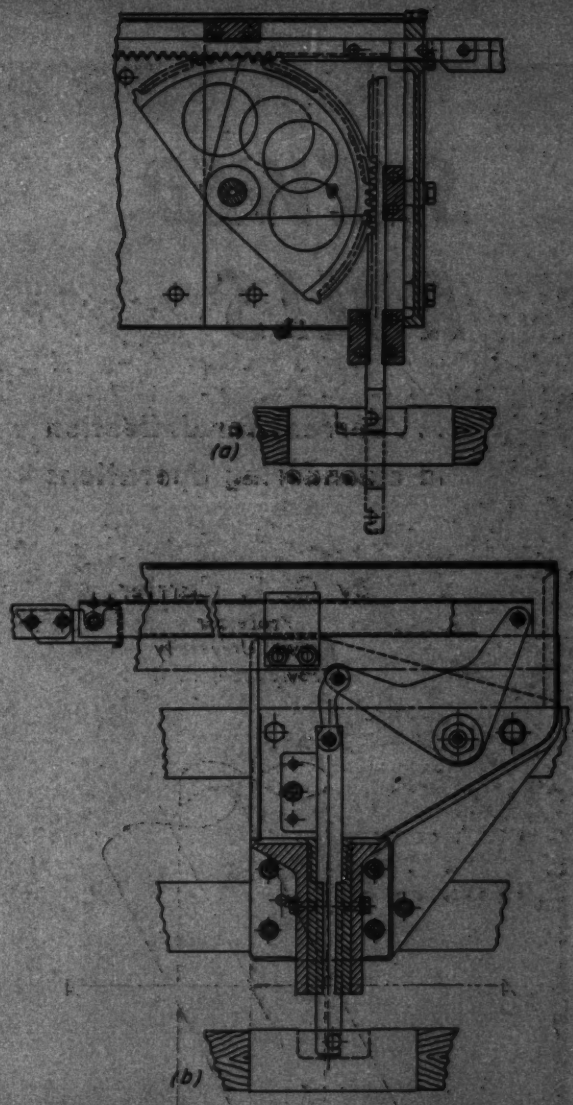
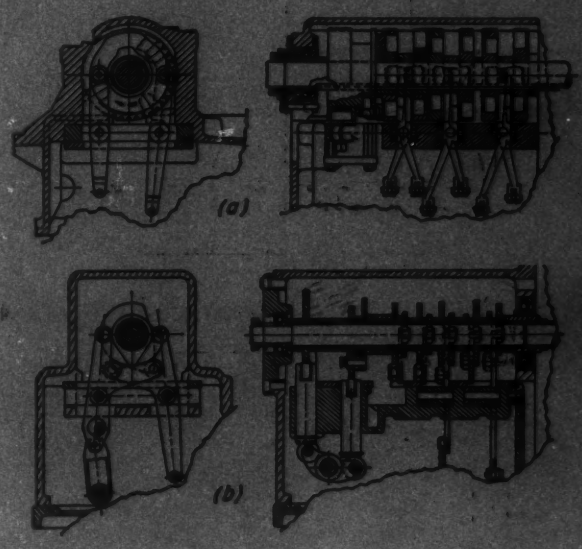


Fig. 5—Above—Right-angle drive (a) redesigned as at (b) resulted in savings in assembly, weight and cost

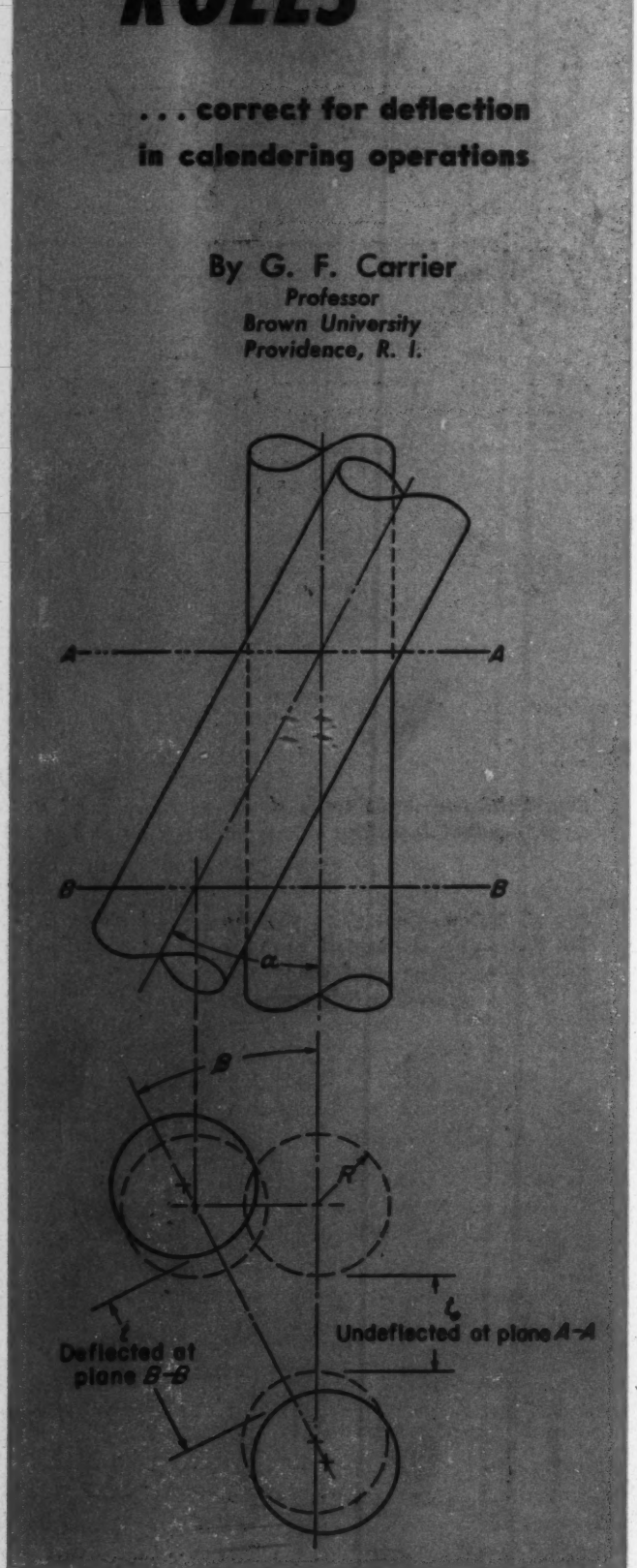
Fig. 6—Below—Cam drive (a) redesigned as at (b) for flat cams eliminated prohibitive cost and heat treating warpage and simplified holding the close operating tolerances necessary



SKEWED ROLLS

. . . correct for deflection in calendering operations

By G. F. Carrier
Professor
Brown University
Providence, R. I.



IN MANY commercial operations, sheets of material are produced by passing a viscous raw material between a pair of rotating rolls. However, when the rolls are supported only at the ends, the uniformity of sheet thickness frequently is lost when large deflections of the rolls occur. These deflections arise, of course, because of the large force resulting from resistance of the viscous material to passage between the rolls.

Since the magnitude of the deflection depends upon the nominal thickness of the sheet being rolled, an adjustable "cure" for this situation is most desirable. Such a cure can be effected by skewing the rolls an amount depending upon the thickness of the sheet and the properties of the material. Although constant thickness is not entirely realized by this correction, mathematical analysis shows that variations in a typical application may be reduced from 20 per cent with parallel rolls to 2 per cent with skewed rolls.

The following equations provide a method of determining sheet thickness and angle of skew. Where t and t_o are the dimensions given in the figure, let

$$s = \frac{t}{t_o}$$

The average and extremes of sheet thickness are

$$s_{avg} = 1 + \frac{43}{45}\lambda$$

$$s_{max} = 1 + \lambda$$

$$s_{min} = 1 + \frac{413}{441}\lambda$$

and angle of skew in radians is

$$\alpha = \frac{\sqrt{4Rt_o\lambda}}{l}$$

where

$$\lambda = \frac{45}{86}\left[\sqrt{1 + \frac{86}{105}\frac{kl^4}{EIt_o}} - 1\right]$$

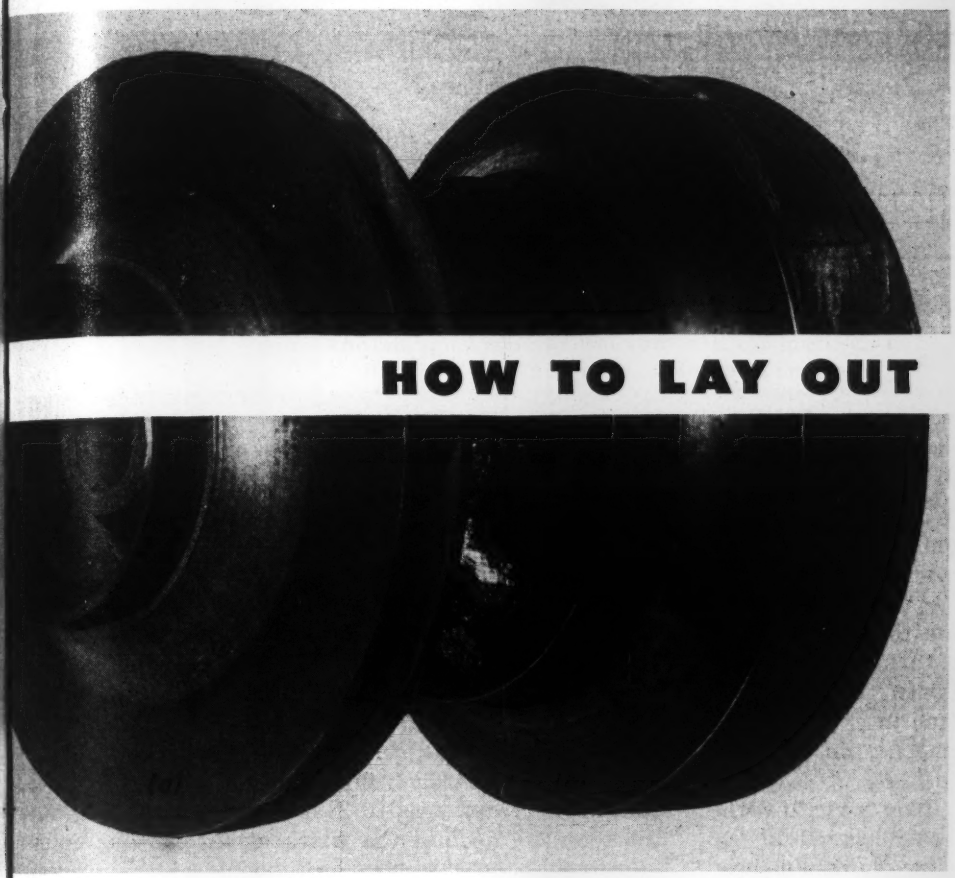
The constant k is defined in the relationship

$$p = \frac{k}{t}$$

where p is the force per unit length normal to the rolls which are considered to be essentially pin-supported beams. The value of k must, in general, be determined from known properties of the material or by experiment. The factors l, E and I are length, modulus of elasticity and moment of inertia of the rolls.

This solution is based on the assumptions that the following quantities are very small: α, β, t_o/R and roll deflection with respect to roll radius.

This article is based on a paper presented at the National Conference of the Applied Mechanics Division of ASME at Purdue University, Lafayette, Ind., June 22-24, 1950.



HOW TO LAY OUT Centrifugal Pump Impellers

By Edward G. Hilficker
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MUCH has been published about centrifugal impeller designs, as well as different methods for developing them. However, detailed information on construction is not so readily available. To remedy this deficiency, the present article sets forth a practical layout method which outlines a step-by-step procedure, so that the designer can develop a straight vane or Francis type impeller, *Fig.* 1, from the figures submitted by the hydraulic engineer.

Usually, design specifications to which the impeller must be tailored include inlet diagram, *Fig. 2a,* and exit diagram, *Fig. 2b.* However, the exit diagram is not always completely shown, *Fig. 2c.* The partial diagram does, however, supply the designer with the required impeller construction data. Along with these diagrams, the quantities D_2, B_2, H_2 and β_2 are usually tabulated (see Nomenclature). These furnish the complete information for the exit end of the impeller. The inlet data may not be as complete, nor will the inlet be as simple to design as the exit, because the vane angle β_1 may have to be adjusted to get enough vane overlap and still give the required inlet height. With a little study and reworking of the inlet end of the vane, efficient results can be obtained. Generally, the D_1, D_h, D_s, and shaft sleeve or impeller nut diameter are specified together with the number of vanes. The hydraulic engineer's figures would also include: flow, Q gpm; total head, H ft; impeller speed, rpm; and casing volute area (throat), sq in.

Fig. 1—Top—Centrifugal impeller types discussed in this article, with straight vane type shown at left and Francis type impeller shown at the right

PROFILE CONSTRUCTION—STRAIGHT TYPE (see *Fig.* 3): To determine the required vane profile, draw the center lines XX and YY. Measure off radial distances for D_s, D_h, D_1 and D_2 and draw horizontal construction lines through these points. Next in order is impeller width, B_2. This, as a general rule, is divided equidistant on either side of the YY axis ($CA = CB$). Drop a vertical construction line from A at D_2 and sketch in curvature to A at D_h. It should be noted that keeping the outer end of line AA perpendicular to the shaft axis will eliminate the necessity of core driers. Some impellers are designed with sloping back shrouds, i.e., narrow at D_2 and widening toward D_1 with respect to center line YY. This is necessary in some designs but should be the exception and not

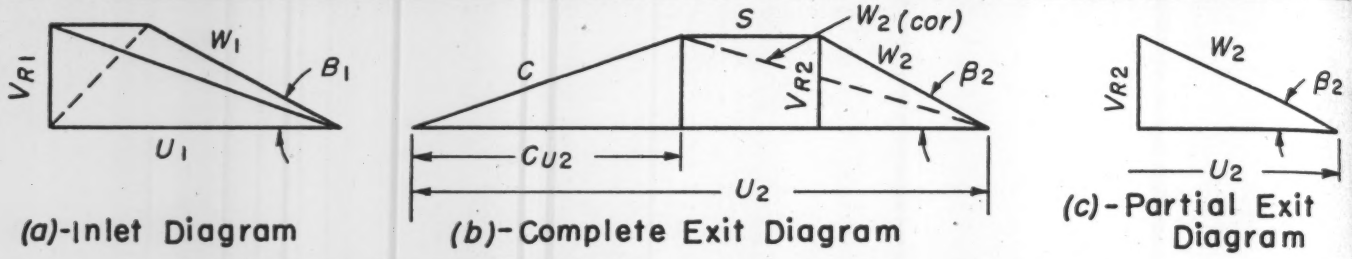


Fig. 2—Impeller design specifications generally include inlet diagram and either complete or partial exit diagram

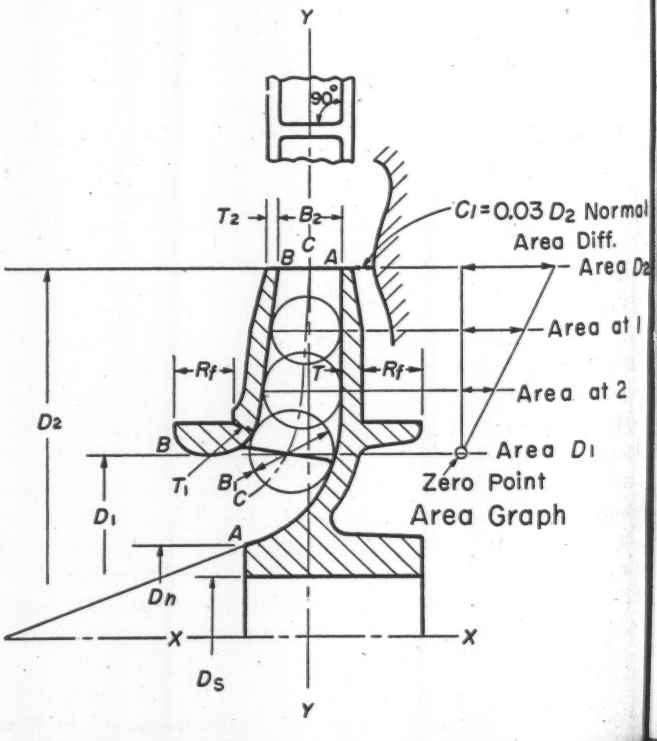
common practice, as the core driers may equal or exceed the initial cost of the impeller pattern.

It is important that a smooth approach be kept in mind when approximating the curvature of line *AA*. The liquid has somewhat of a spin in the impeller eye and as it reaches the impeller vane the direction of the liquid is changing. In order to allow for this change in direction, the ratio between the eye area and inlet radial area, neglecting vane interference, is made greater by 1.3; i.e., area at impeller inlet = 1.3 [D_1 (area) − D_h (area)] or width B_1 = 1.3 [D_1 (area) − D_h (area)]/πD_1. Inscribing a circle with diameter equal to B_1, having its center located on D_1 and tangent to the approximated line *A*, establishes the width at the inlet.

The next step is to draw the area graph in order to control the profile widths between D_1 and D_2. Calculate the areas, neglecting vane interference, at both of these diameters (area = $\pi D \times$ width). Subtracting the lesser from the greater, set up the triangle with the lesser area representing the zero point, *Fig.* 3. Draw a vertical line from the zero point through the horizontal line representing the larger area, and lay off the difference at the other diameter. Graduate the distance between these two points on the vertical line. These intermediate lines will represent diameters, *Fig.* 3. Measure the distance from the vertical line to the diagonal, using the same scale as used for the area difference. Adding this additional area to the smaller area results in a new area at a specific diameter. Calculate the width at the various diameters using the corrected area, using width = area/πD. Then draw a circle on each graduate line, tangent to line *A*, using the calculated width as a diameter. Connect each circle with a tangent from *B* on D_2 to *B* at D_1. Care should be exercised in determining the

Nomenclature

D_2 = Impeller outside diameter
D_1 = Impeller eye diameter
D_h = Impeller hub diameter
D_0 = Impeller inlet tip (mean flow line)
D_s = Shaft diameter at impeller
B_1 = Impeller inlet width
B_2 = Impeller outlet width
H_1 = Impeller inlet height
H_2 = Impeller outlet height
β_1 = Impeller inlet angle
β_2 = Impeller outlet angle
T = Shroud thickness
T_1 = Shroud thickness at ring turn
T_2 = Shroud tip thickness
R_f = Ring finish length
C_l = Impeller-to-casing clearance
W_1 = Relative inlet velocity
W_2 = Relative exit velocity
U_1 = Peripheral speed at D_1
U_2 = Peripheral speed at D_2
V_{r1} = Radial inlet velocity
V_{r2} = Radial exit velocity
C = Absolute velocity
C_{u2} = Tangential component of absolute velocity
α = Gallons per minute
Q = Vane diffusion angle



shape near D_1 so that at no time will the hydraulic passage between the shrouds *A* and *B* digress from either parallel walls or a constant decrease in width, as viewed from D_1 to D_2.

The hydraulic passage having been completed, the ring diameter and length should be determined with consideration given to type of service, future applications and space limitations. It should be understood that with normal running clearances of approximately 0.008 to 0.012-in., the smaller the ring diameter the less is the possibility of leakage.

VANE CONSTRUCTION—STRAIGHT TYPE (see *Fig.* 4): With the completion of the profile, the vane development is the next step. A logarithmic spiral (straight) type vane, which is probably the most common in centrifugal pump design, will be developed here. The term "straight vane" does not means a straight radial blade but refers to the vane and its relation to both front and rear shroud, lines *BB* and *AA*, respectively, *Fig.* 3. The angle between the vane and shrouds is consistently 90 degrees for the entire length of the "straight vane".

Drawing Grid Development

The mean flow line *CC*, *Fig.* 4*b*, is used for the vane and grid development. The length of line *CC* is stretched out in a straight line *C'C'*, *Fig.* 4*c*, and marked off in equal spaces as indicated at 0, 1, 2, 3 and 4 with the last space, 4-*e*, (end of vane) being the odd space. After drawing the horizontal lines 0, 1, 2, etc., for the grid development, the vane can be drawn using as a starting point the exit angle β_2. If the inlet angle happens to be the same magnitude as the exit angle, the upper side of the vane will be a straight line. If the angles β_2 and β_1 differ, start the grid development by drawing the upper side of vane as if both angles were the same. Set the drafting machine for the specified inlet angle and approximate the tangent point of H_2. From this point draw the inlet angle and smooth the hump with a long curve. Care must be exercised so that all changes of direction are gradual.

Project the spacings 0, 1, 2, etc., from profile line *CC* on *Fig.* 4*b* over to line *Y'Y'* (*Fig.* 4*a*) and draw arcs through each point. Now transfer the distances L_0, L_1, L_2, etc., from *Fig.* 4*c* on to *Fig.* 4*a* as indicated by L_{0R}, L_{1R}, L_{2R}, etc., always dropping a radial line from the point indicated to the next inner circle, e g., step off a distance on the circle 0 from the *XX* axis L_0 and drop a radial line from this point on circle 0 to circle 1, and so on until all vane points are marked. Connect these intersections with a spiral curve and this will be the upper side of the vane.

Next, approximate a vane thickness and sketch in the lower side of the vane, rounding the inlet edge as shown. Having done this, make a template of the vane and swing the template to the second vane position and observe the hydraulic passage, *Fig.* 4*a*. If openings are nearly correct

$$H_1 = \frac{Q \times 0.321}{W_1 \text{ (No. vanes)}} \quad \ldots\ldots (1)$$

$$H_2 = \frac{Q \times 0.321}{W_2 \text{ (No. vanes)}} \quad \ldots\ldots (2)$$

and if the passage is gradually spreading from H_1

Fig. 3—Left—Method of constructing straight type impeller profile

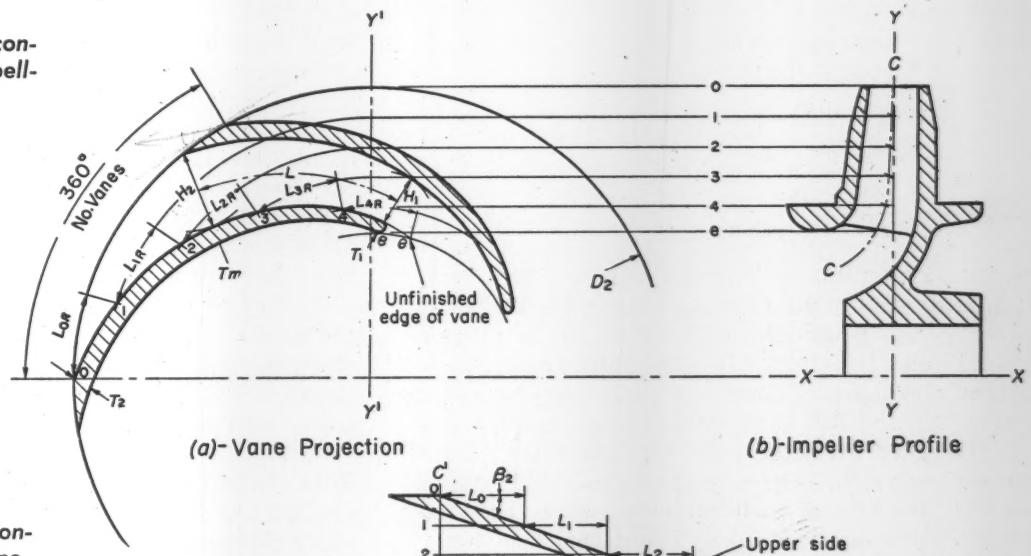
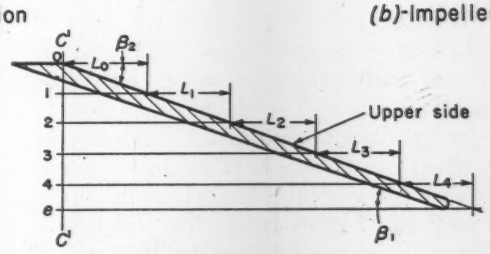


Fig. 4—Right—Method of constructing straight type vane



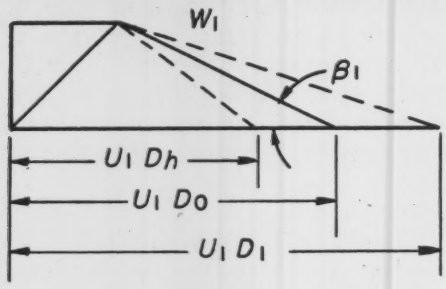
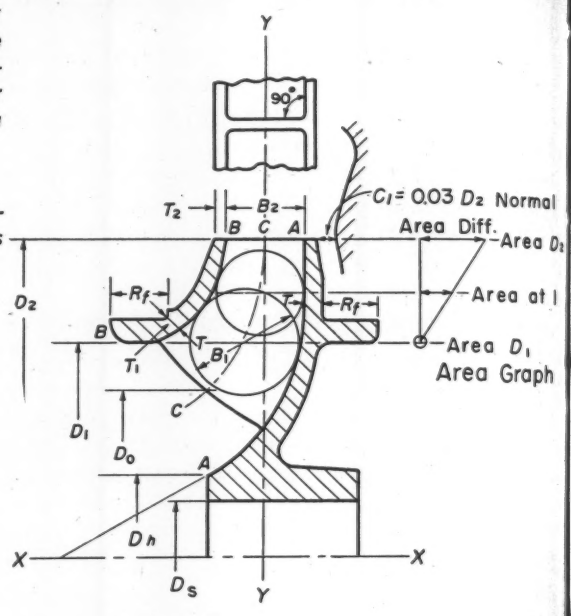


Fig. 5—Left—Inlet diagram for Francis impeller showing three inlet vane tip angles. Compare with single angle for straight vane type, Fig. 2a

Fig. 6—Right—Method of constructing profile for Francis type impeller



to H_2, *Fig.* 4*a*, the vane development is finished. However, if the openings are not correct a slight change in the position of the inlet tip will make considerable change to the hydraulic passage. Maintain the same angles but move the point of contact with the exit angle line. Moving the inlet tip to the left will enlarge the openings while moving it to the right will decrease the openings. If the spread between the vanes is unsatisfactory, try an additional vane, or one less. Sometimes this will help to adjust the port.

Checking Diffusion Angle

Always check the spread between H_1 and H_2 by means of tangents. This "diffusion" angle is given by the relation

$$\frac{H_2 - H_1}{L} = \text{tangent } \alpha \quad \ldots\ldots (3)$$

A good vane diffusion angle should be about 8 to 11 degrees. Under no circumstances should it exceed 14 degrees.

Before the drawing is finally approved, the velocities W_2 and W_1 should be checked

$$W_2 = \frac{Q \times 0.321}{H_2 \times B_2 \times \text{No. vanes}} \quad \ldots\ldots (4)$$

$$W_1 = \frac{Q \times 0.321}{H_1 \times B_1 \times \text{No. vanes}} \quad \ldots\ldots (5)$$

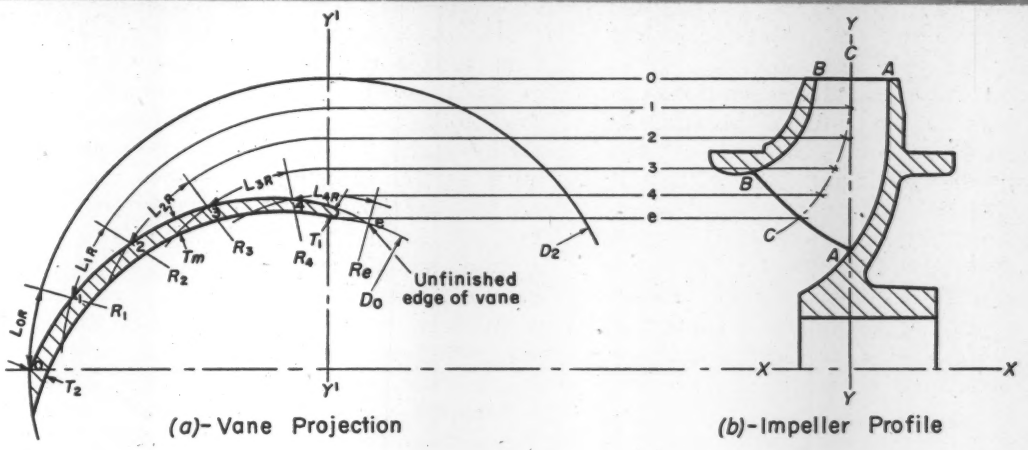
PROFILE CONSTRUCTION—FRANCIS TYPE: This vane design is commonly used for the higher specific speed type of pump, where the ratio of D_2/D_1 is greater than 1.25 and less than 2.0. The inlet tip angle β_1 constantly changes between the inner and the outer shrouds, *AA* and *BB*. The smaller angle corresponds to the smaller diameter near D_h, while the larger angle occurs at D_1. Some designers use this type of vane with the idea of maintaining a rising "head capacity" characteristic curve. The author has found that this is not always true, although the tendency is generally in that direction. There is of necessity more development to this type of vane but there is little difference in the method of procedure. The inlet diagram, *Fig.* 5, shows three angles for the inlet vane tip as compared with one for a straight vane, *Fig.* 2*a*. Otherwise the hydraulic figures will be identical to those listed for the straight type vane. The profile construction, *Fig.* 6, for the Francis type is the same as for the straight type vane previously given.

VANE CONSTRUCTION—FRANCIS TYPE (see *Fig.* 7): Contrary to the straight vane, the Francis type vane has a different angle with the shrouds at the inlet tip and sometimes does not reach a 90 degree angle with its shroud at any place along its entire length. This of necessity means a twisted vane between inlet and exit tip. In some designs the length of the twisted section is very short; therefore, the inner and outer faces of the vane coincide and result in a 90 degree angle with the shroud. *Fig.* 8 shows a twist as far as D_2, where the vane faces coincide. This means that the 90-degree angle is reached only at the exit tips and is changing between inlet and exit.

Proceed with the vane and grid development, *Fig.* 7, as for the straight vane construction, using the mean flow line development.

Having developed the vane upper side along the mean flow line, the next step would be to approximate the thickness and complete the development as shown in *Fig.* 7.

In order to establish the twisted section of the vane it is necessary to study the profile carefully. Referring to *Fig.* 8*b*, point *P* is located by observation and may be relocated to improve vane development. However, the sections R_1, R_2, R_3, R_4 and R_e are radial sections and best results will be obtained if the angularity between the inner and outer shroud is divided as equally as possible along the entire length of the twisted section. Holding a point at some position outside the profile and swinging a triangle between the inlet tip and exit diameter, a fairly good location for *P* can be determined. Lines drawn through *P* and 1, 2, 3, 4 and *e* on line *CC* will



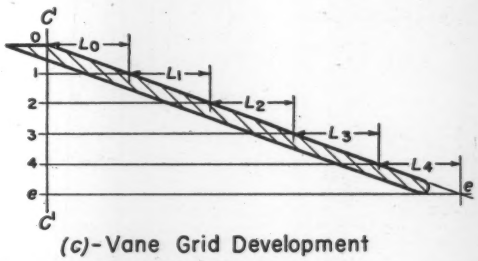


Fig. 7—Right—Construction of Francis impeller vane, similar to Fig. 4 for straight type

Fig. 8 — Below — Determining vane surfaces of Francis impeller

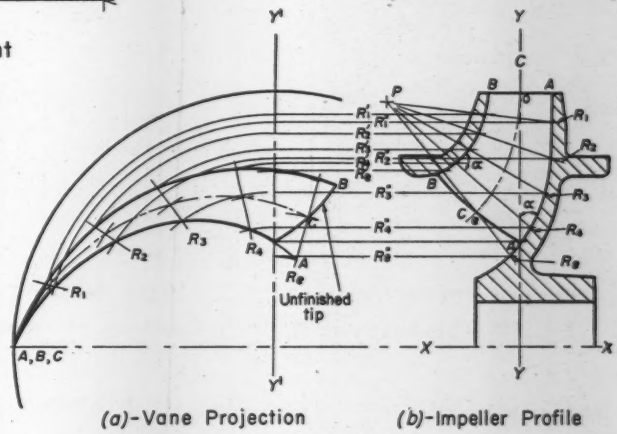
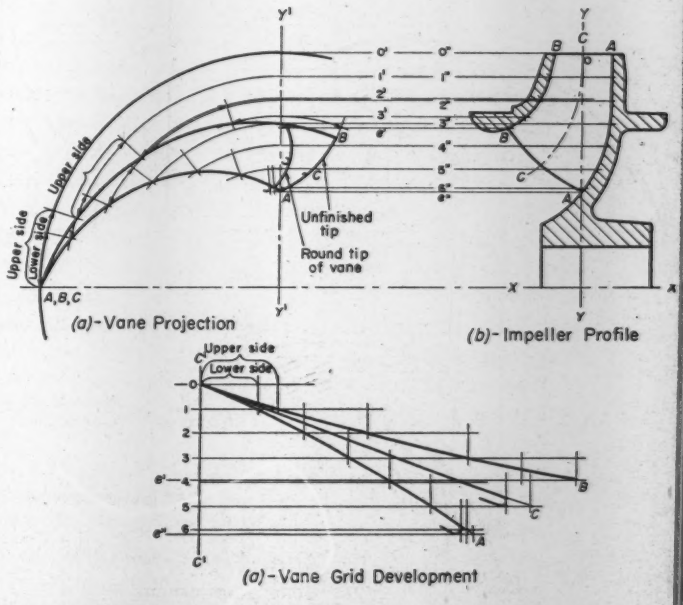


Fig. 9—Below—Reverse layout process to check vane edges for smoothness



set up radial sections across the vane upper face from which the remainder of the vane can be developed. Because of this, it is important to be sure that point *P* is in the best possible location.

The next step is to project the inner and outer edge of the vane onto the vane projection, *Fig. 8a*, R_1', R_2', etc., representing the outer edge, while R_1'', R_2'', etc., represent the inner vane. Where R_1' and R_1'' intersect the radial line R_1, on vane projection *Fig. 8a*, indicates the extreme edges of the vane where it contacts the shroud. A curve drawn through these points will indicate the complete upper surface of one vane.

Checking Vanes for Smoothness

In order to insure smooth lines for both edges of the vane, proceed with the reverse process as shown in *Fig. 9*. First step off distances on lines *AA* and *BB*, *Fig. 9b*, using the same increments as used on *CC* and leaving the last space to be the odd one. Then project points 1′, 2′, 3′, etc., on the outer edge of the vane, (line *BB*) over to line *Y′Y′*, *Fig. 9a*. Setting the compass to a radius that intersects each point, draw the radii through the *BB* edge of the vane. Do the same with points 1″, 2″, 3″, etc., but draw these radii through the *AA* edge of the vane. Set either compass or dividers to the chord lengths between successive points on lines *AA* and *BB*, *Fig. 9a*, measured on arcs having their center at the *XX-Y′Y′* intersection. Step off these distances on the grid development, *Fig. 9c*, both lower and upper sides, the lower side being the shorter, or steeper angle. Follow through for each point and draw a line connecting each point on the grid. If there are

no sudden changes in contour, a smooth surface for the vanes is assured. The more accurate the projection and curves connecting the points, the less troublesome will be the grid development.

The next step is to check the flow conditions between two vanes. Make a template of the vane as shown in *Figs.* 7 and 8 and rotate the template the proper angle and copy the second vane. Check the exit height, Equation 2, and the exit velocity, Equation 4. Also check the angle of deflection between the vanes from Equation 3. This should be about 8 to 11 degrees, but never exceed 14 degrees maximum. Should the openings not appear as expected, a change in position of the inlet tip will be necessary, as previously discussed. If a change in the tip is made, the development of vane and grid will of necessity have to be repeated.

The patternmaker, as a rule, requires board sections to construct the vane. Therefore, to allow him to prick punch the section through to his boards, the profile and vane development is drawn to a shrink scale. These board sections and a representative pattern drawing are shown in *Fig.* 10. Measure off the board sections beginning on the right, *Fig.* 10*b*, so that the odd board will be at the shorter section, or left-hand vane tip. Draw vertical lines between shaft centerline D_2 of the impeller. Set a compass to $r_{1\text{-}4}$ intersections of board section 1 and radial section R_4. Intersect radial section line R_4 on vane projection, *Fig.* 10*a*. Continue with intersections $r_{1\text{-}3}$, $r_{1\text{-}2}$, and $r_{1\text{-}1}$. Draw a curve through these intersections on radial lines R_4, R_3, R_2 and R_1, meeting on inner edge of vane at point 1. This is board section 1. Follow the same procedure for the other board section and the finished drawing will look similar to that shown in *Fig.* 10.

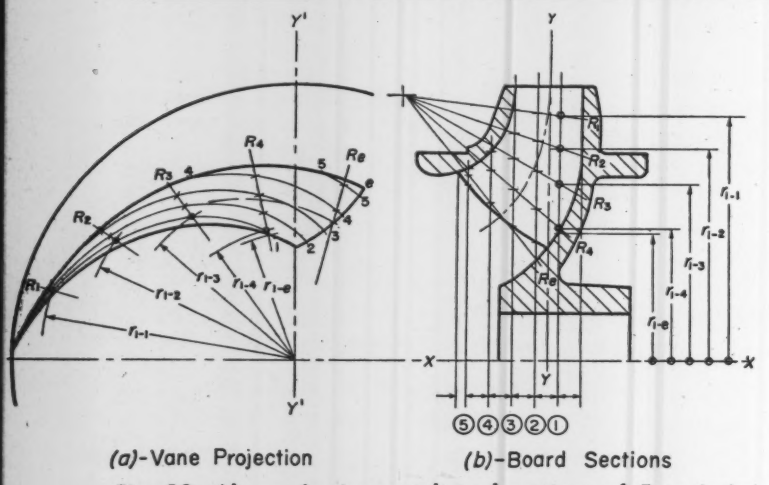
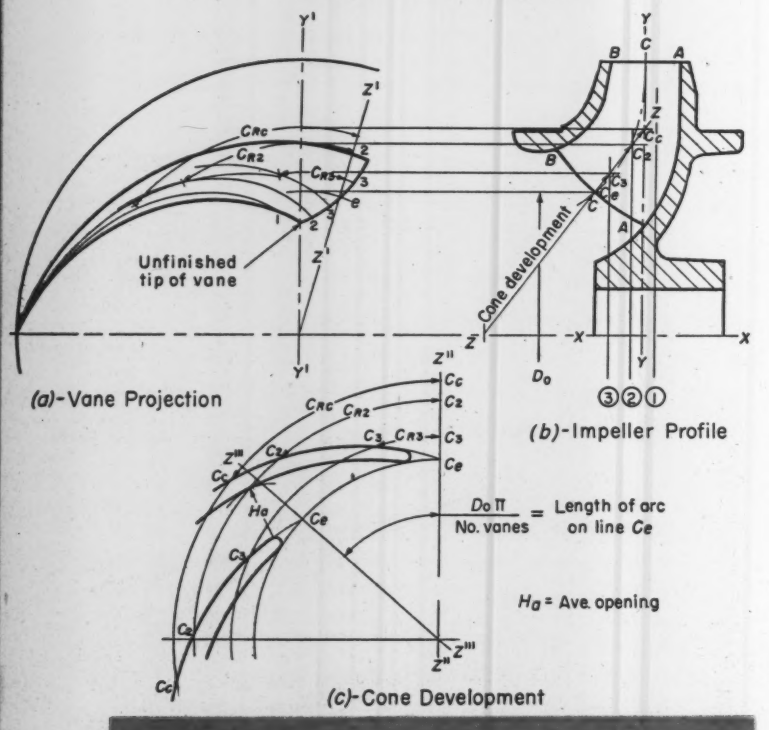


Fig. 10—Above—Laying out board sections of Francis impeller for patternmaker

Fig. 11—Below—Cone development for determining the inlet height of Francis impeller


Unfinished tip of vane
Cone development
(a)-Vane Projection
(b)-Impeller Profile
Do π / No. vanes = Length of arc on line Ce
Ha = Ave. opening
(c)-Cone Development

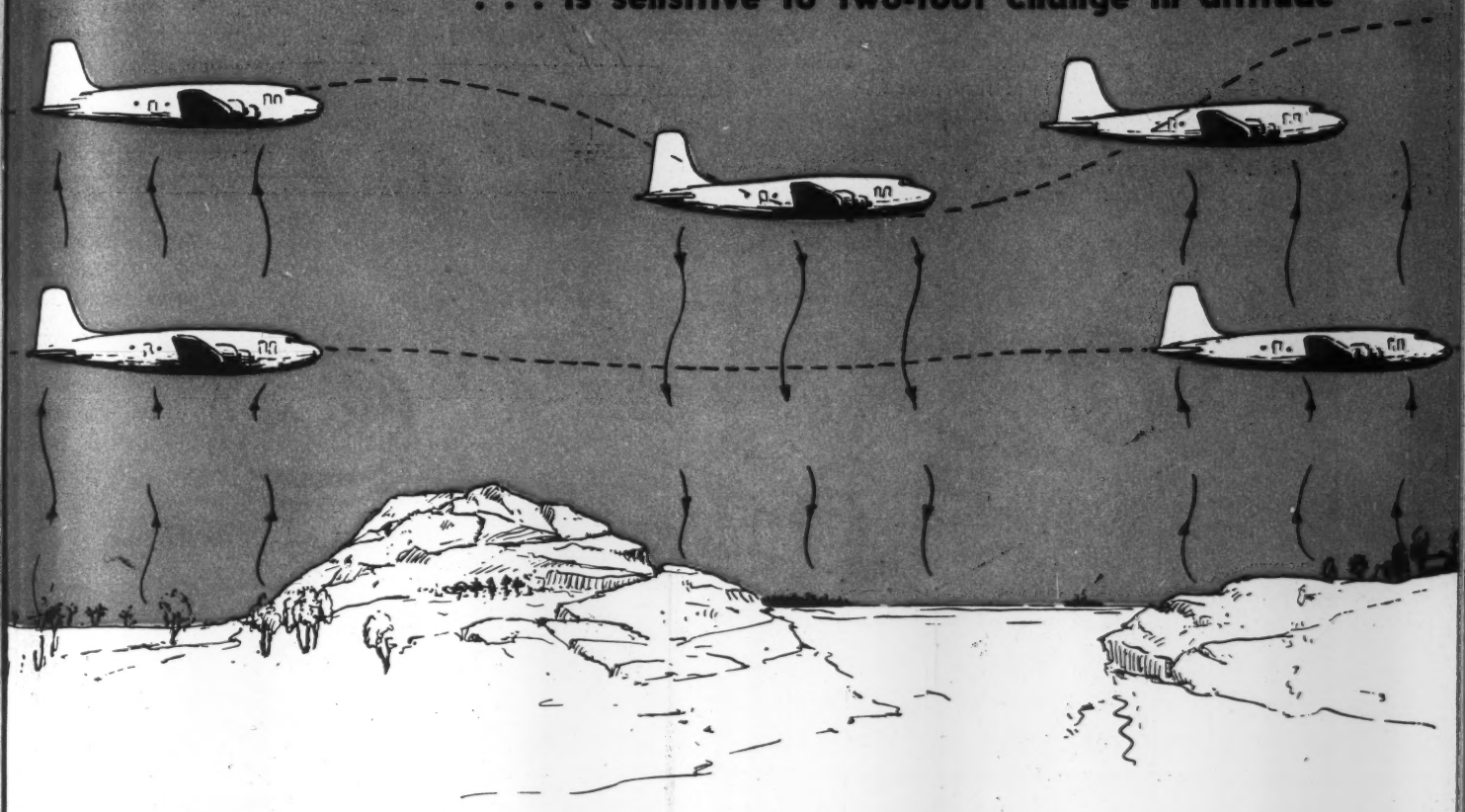
Cone Development Sets Inlet Height

The last step is a cone development for establishing the the inlet height H_1 as shown in *Fig.* 11. As the vane angle is continually changing on the tip from *A* to *B*, the mean or average opening will occur at the meridian flow line *C*. Draw line *ZZ*, *Fig.* 11*b*, approximating the nearest approach to line *CC*. It is not possible to have line *ZZ* coincide with *CC* but it should not be too far afield. Having established line *ZZ*, describe, on line *Z″Z″*, *Fig.* 11*c*, arcs equal to lengths of cone development lines C_e, C_3, C_2 and C_c from *Fig.* 11*b*. Then calculate the distance between vanes, setting down *Z‴Z‴* for the second vane on *Fig.* 11*c*. On vane development, *Fig.* 11*a*, locate line *Z′Z′* extended from the center through the intersection of *CC* with unfinished tip of vane and project points C_e, C_3, C_2, C_c, over to *Y′Y′* on the vane projection, *Fig.* 11*a*. Draw arcs through these points and intersect the board sections passed through on the vane profile. Measure the arcs C_{R3}, C_{R2}, C_{Rc} on *Fig.* 11*a* and step off on the cone development, *Fig.* 11*c*, beginning at C_e. Draw a curve through the intersections C_e, C_3, C_2, C_c for two vanes and, after approximating the vane thickness, measure inlet height H_1. If the port opens gradually from inlet to exit, check the inlet relative velocity from Equation 5. If this relative inlet velocity is satisfactory, the development is complete. The working drawing can be a tracing of the developed impeller with necessary dimensions for machining. The patternmaker should be furnished with a vane development similar to *Fig.* 10, and a print of the working drawing.

This method of vane development gives the designer control of the vane for its entire length. It insures against sudden changes in contour and guarantees the required openings. Drawing is simplified, especially when establishing the board sections and outer and inner edges of the Francis type vane. The drawing time is considerably shortened when once the designer becomes familiar with the mechanics of the method.

Flight Control Unit

. . . is sensitive to two-foot change in altitude



By Michael A. Argentieri
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DESIGNED to control the altitude of an aircraft in conjunction with an electronic autopilot, a new type of flight unit, sensitive to a two-foot change in elevation, maintains constant flight as indicated schematically in *Fig.* 1. This altitude controller utilizes an aneroid-actuated mechanism coupled to an inductive pickoff through a direct-current magnetic clutch. Gyros, the prime controls of an autopilot, are capable of sensing the variations in an aircraft's attitude known as pitch, roll, and yaw. Unless accompanied by a change in attitude, however, they cannot sense vertical ascents or descents of an aircraft such as are caused by improper trim or vertical gusts, commonly referred to as up or down drafts. This controller, therefore, provides an important added "sense" to present-day autopilot control by overriding the gyros to the extent necessary to maintain the aircraft at a fixed barometric altitude.

Fig. 1—Above—Comparison of aircraft flight when controlled by autopilot alone as shown in upper flight curve, and when altitude controller is employed in conjunction with an autopilot, lower curve. Although attitude of the plane during the former flight does not change, its altitude does due to up and down drafts. The controller maintains flight level within 10 feet

The use of automatic altitude control lessens pilot fatigue by simplifying the mechanics of flight, thereby freeing the pilot so that he can focus his attention on other important duties. It is particularly helpful under instrument conditions where air traffic laminar control is essential for safety.

In military aircraft, altitude control is desirable to help provide a more stable platform for precision bombing and fire control.

The portion of the controller containing the clutch and inductive sensing pickoff is also applicable to instruments measuring other flight functions such as the control of air speed, Mach number, and angle of attack.

PRINCIPLES OF OPERATION: Three basic components of the controller, shown in *Fig.* 2 are identified as:

Aneroid Diaphragms: The controller's prime mover consists of two aneroid diaphragms which expand with increasing altitude due to lessened air pressure. They are arranged as a couple to produce an angular motion of a shaft. This arrangement serves to bal-

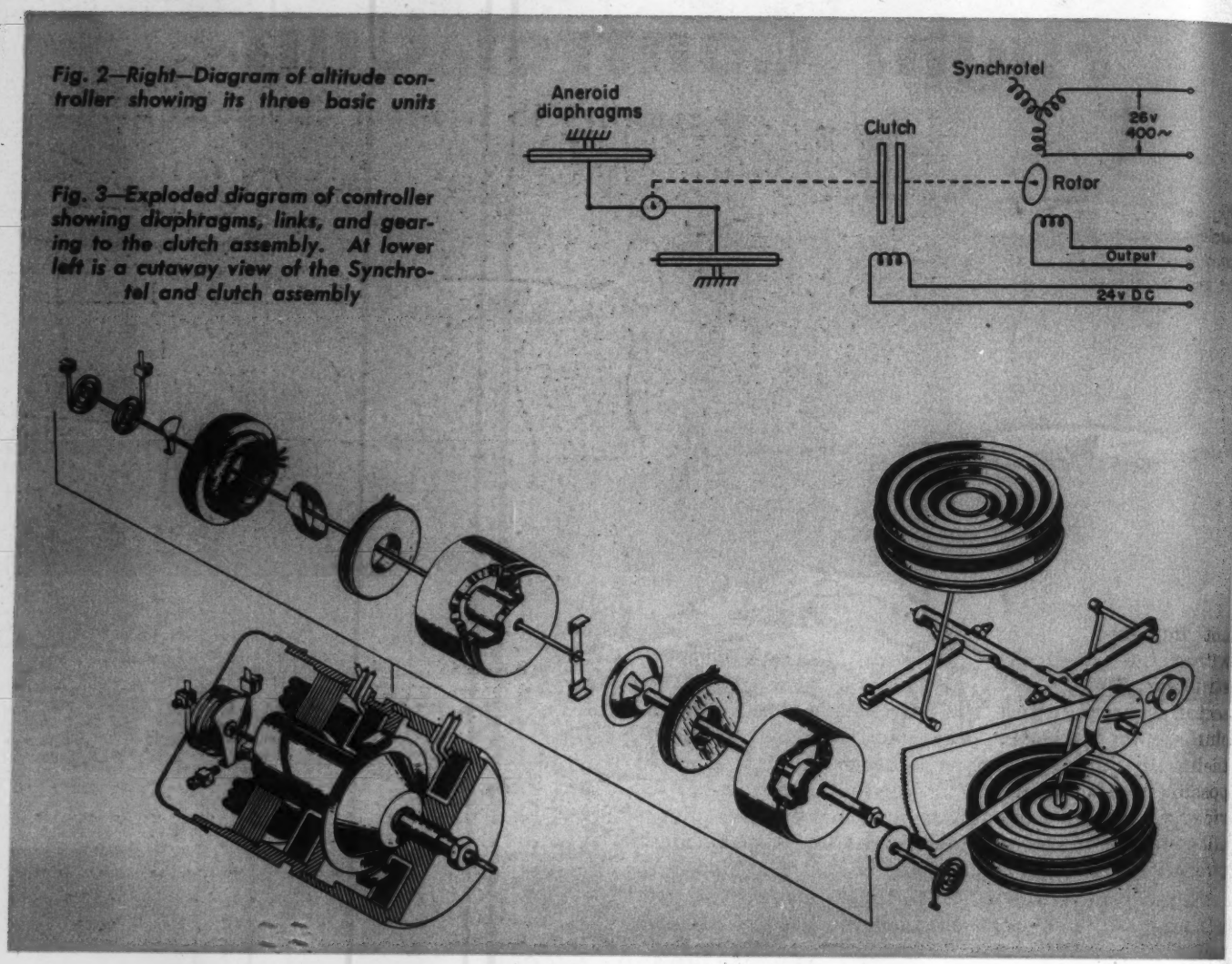


Fig. 2—Right—Diagram of altitude controller showing its three basic units

Fig. 3—Exploded diagram of controller showing diaphragms, links, and gearing to the clutch assembly. At lower left is a cutaway view of the Synchrotel and clutch assembly

ance the diaphragms, virtually eliminating errors due to accelerations, position, and *g*-load of the aircraft. Temperature compensation is accomplished by a bimetal bracket surrounding the diaphragm.

Magnetic Clutch: An electromagnetic clutch couples the output of the aneroid diaphragms to the rotor of an inductive sensing unit, *Fig.* 3, in this case a Kollsman Synchrotel. The application of a direct-current voltage across the clutch coil sets up a magnetic field which attracts to the clutch disk two spring-mounted iron armatures fixed to the Synchrotel rotor shaft.

Inductive Sensing Pickoff: The Synchrotel has its rotor spring centered so as to give a null voltage between the two output leads when the input coil is supplied with 400-cycle alternating current. Any variation in altitude after the clutch has engaged will cause the diaphragms to move the rotor of the Synchrotel. This gives an output voltage amplitude proportional to the sine of the angle away from the null, and a phase reversal of 180 degrees. The voltage is then combined in the autopilot with signals from the gyros and control surface, and amplified to operate a servomotor which controls the position of the aircraft's elevators.

Sensitive to a two-foot change in altitude the instrument permits a well designed autopilot to maintain an aircraft in level flight within ± 10 ft. The pilot can, therefore, call upon the controller to maintain the altitude of his craft at any level up to 50,000 ft by simply snapping a switch to energize the magnetic clutch. The controller itself may be located in any part of the aircraft, a plumbing connection to the static pressure system and a wire cable to the autopilot amplifier being the limiting considerations.

DESIGN FEATURES: The simple frame design can be seen in *Fig.* 4. Four square vertical posts are press fitted and staked into the main face plate. They serve to clamp the aneroid diaphragms and rocking shaft in their relative positions together with a top plate, which supports the clutch and inductive pickoff assembly. The vertical posts are drilled and reamed, then slotted with a circular saw. By tightening the screw located at right angles to the saw slot, the diaphragm post is securely clamped in position. This construction permits a stepless adjustment in diaphragm position.

Sapphire Bearings Increase Sensitivity

Adjustable Gear Mesh: The low friction level and high sensitivity of the mechanism is assured by the exclusive use of sapphire jewel bearings and single-stage gearing with an adjustable gear mesh. In a built-up frame construction it is most difficult, if

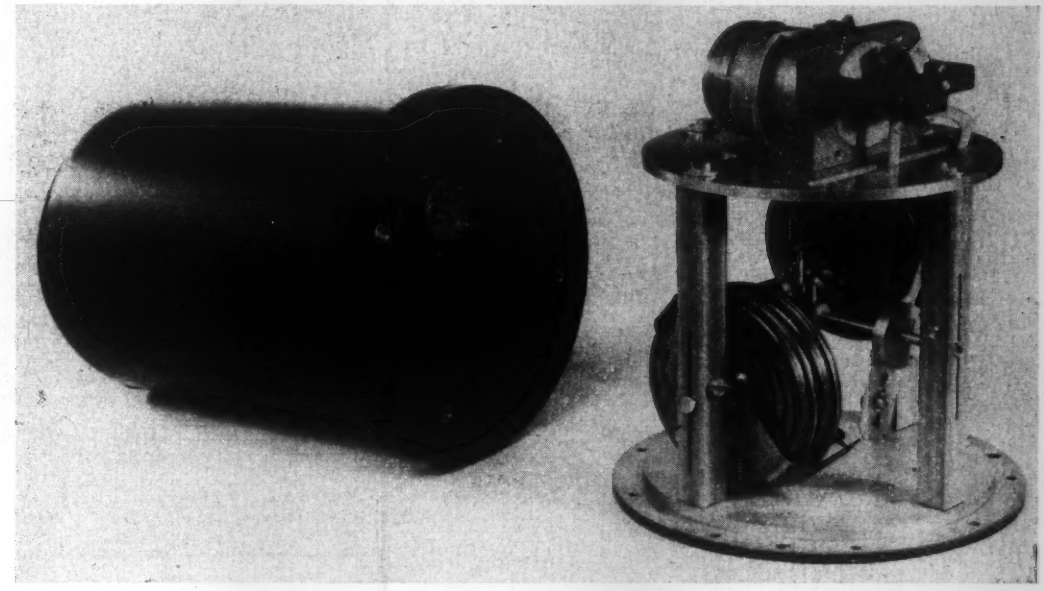


Fig. 4 — Complete altitude controller at left and with cover removed at right, showing frame construction

not impossible, to maintain gear center distances within the tolerances required in so precise an instrument. Therefore, the rocking shaft, to which the sector is clamped, is mounted in two eccentric sapphire jewel bearings. Eccentricity of the jewels in their housings amounts to as much as the largest possible tolerance build-up. A screw driver slot is provided in each jewel housing by which it may be turned, allowing the sector to move toward or away from the pinion gear with which it meshes.

Antibacklash Gearing: A hairspring mounted on a special shaft which is geared back from the main pinion removes backlash from the gear mesh. To facilitate the adjustment of the proper gear mesh between the main pinion and the hairspring idler gear, the hairspring bridge support can be rotated.

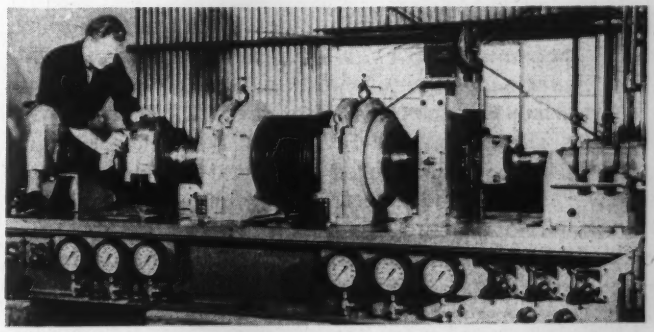
Clutch Airgap Adjustment: The air gap between the main clutch face and the two spring-mounted armature pieces must be controlled to insure the operation of the clutch within the limits of applied voltage. The main clutch disk assembly rides on the threaded portion of the pinion shaft and can be rotated into its proper position. With correct air gap setting, the magnetic clutch will hold a torque of 25 gm-cm, equivalent to a holding range in altitude of $\pm$3000 ft. This torque is much greater than ordinarily required in practice. Since the control will always tend to keep altitude constant, the Synchrotel centering spring will always be near its neutral position where no torque is required of the clutch.

Synchrotel Rotor: The Synchrotel rotor assembly weighs 0.76 grams and has a moment of inertia of 0.11 gm-cm^2. The absence of brushes and the fact that the rotor pivots rotate in sapphire jewel bearings gives a static friction torque of less than 0.005 gm-cm, requiring a negligible positioning torque.

Evaluates Performance of Bearing Lubricants

RAILROAD car journal-box bearing lubricants can be tested on a new machine that simulates actual operating conditions. The tester permits studies of bearing lubrication under shock loading conditions which were previously impossible in the laboratory.

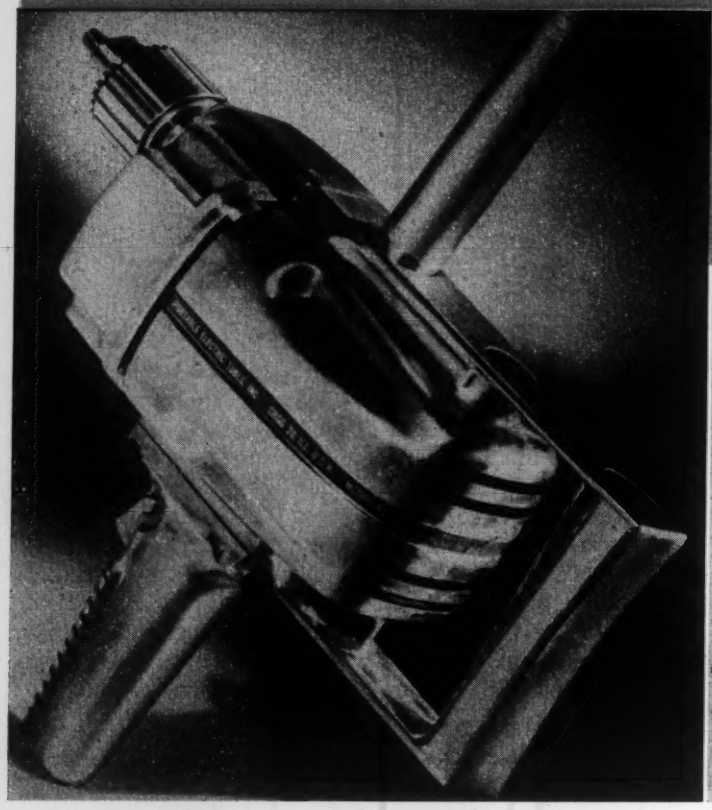
Simulating car speeds up to more than 100 miles per hour, the apparatus duplicates both steady and shock loads encountered in actual railroad operations. Bearings of any manufacturer can be accommodated and the machine is constructed in conformance with American Association of Railroads standards of axle design. It is possible to apply a combined vertical load of 50,000 lb and an axial load of 15,000 lb on each of the two test bearings, one located on each end of the machine. All loading is accomplished hydraulically, and each of the two test bearings can be loaded independently. Designed by the Beacon, N. Y., laboratories of The Texas Co., the machine measures bearing friction, load, temperature, and speed. To minimize vibration and noise, it is built on concrete piers sunk six feet into the ground.

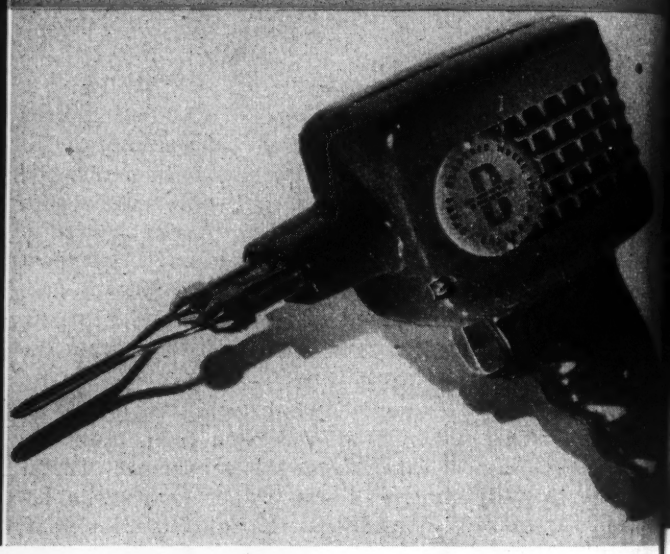


STYLING TRENDS

. . . favor honesty and simplicity in shapes

By David Painter
Vice-President in Charge of Design
Barnes & Reinecke Inc.
Chicago, Ill.





IT IS easy to recall when streamlining was a synonym for styling. Fortunately, that era has become history. Although streamlined designs were often more offensive than those they replaced, the fad brought about a minor revolution in designing for appearance. Styling has again become a more subtle process. Today a streamlined product is as dated as the Empress Eugenie hat, except where the shape is justified by function. Thanks to the streamlining vogue, however, management is appearance conscious and acknowledges the need for styling which today tends toward a clean and direct approach to shapes and forms. This current trend is exemplified by the accompanying illustrations.

Contrast the portable electric drill, *Fig.* 1, with similar equipment of a few years ago. The housing of this drill, manufactured by Portable Electric Tools Inc., is concise and direct. The curves are in the interest of condensing the bulk to its minimum, defining a well planned shape that eliminates costly finishing operations. Aluminum die casting was selected for the housing because mountings and locating bosses to support the mechanical and electrical parts could be designed as integral parts of the castings. The castings are brush-finished natural aluminum and are assembled with exposed screws.

Reflecting the same direct approach, the electric soldering iron, *Fig.* 2, manufactured by Drake Electric Works Inc., utilizes a molded phenolic case. The shapes are designed to accent the compactness desirable in such tools and the handle has a pitch to provide ease of operation and comfort in use. Phenolic plastic was selected for its light weight and durability. Ventilating louvre areas are flush with the case itself to minimize chipping and breaking of projecting edges and the cowl provides protection for the work lamp. Color of the material and ease of assembly, through elimination of extra parts, are other factors that influenced the selection of phenolic plastic. The nameplates inexpensively achieve accent with color contrast and become focal points in the design.

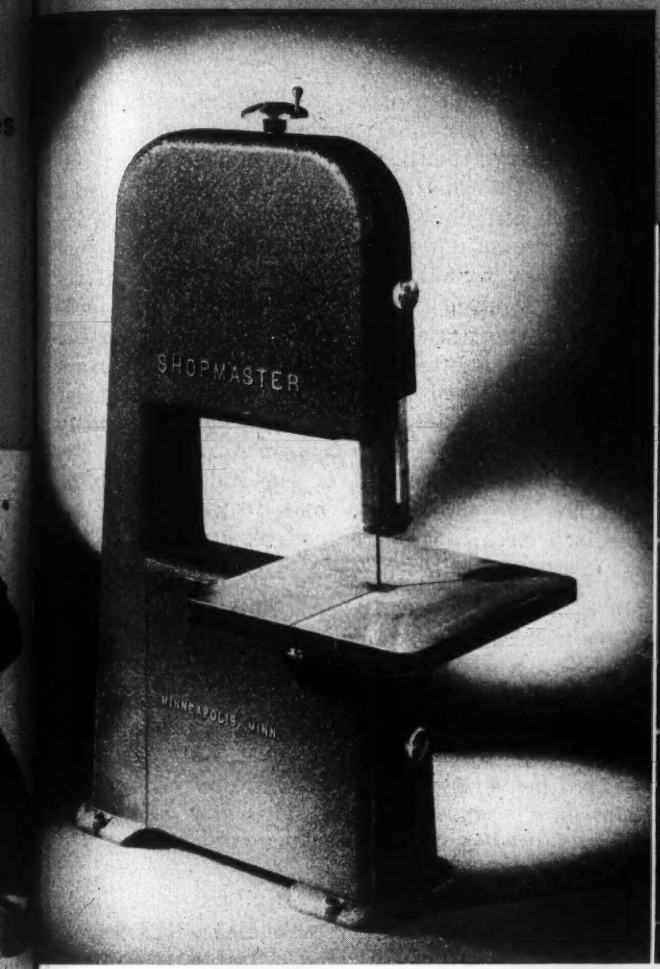
Shopmaster Inc. has employed similar simple styling for the home workshop band saw, *Fig.* 3. Honesty

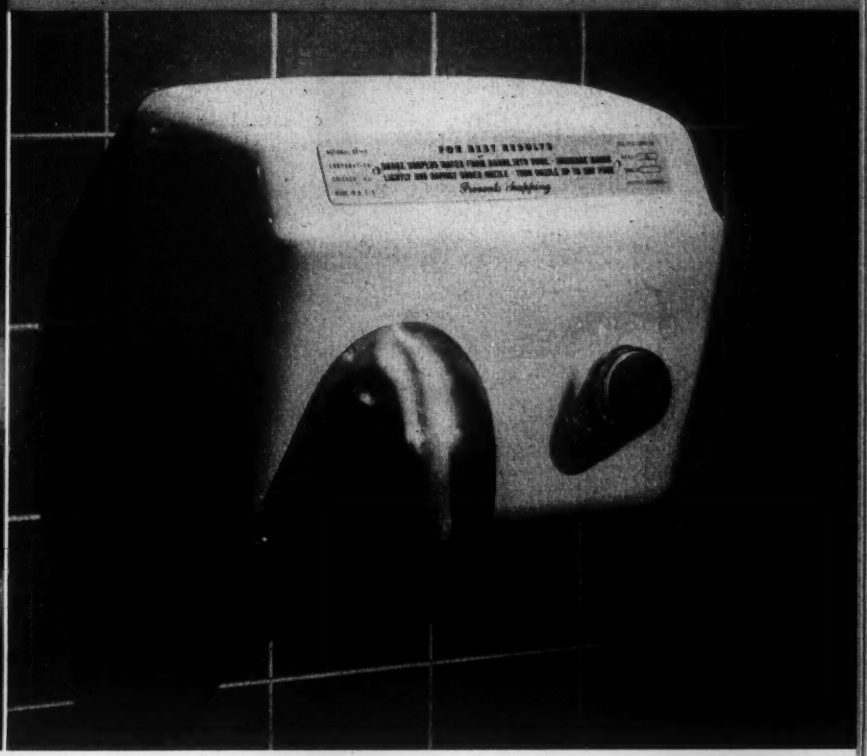
Fig. 1—Extreme left—Portable electric drill designed to keep size and weight to a minimum and eliminate costly finishing operations

Fig. 2—Left, center—Molded phenolic case on soldering gun is attractive in color and easy to assemble

Fig. 3—Left—Simple styling of band saw utilizes aluminum castings for frame, simplifying construction by obviating sheet metal enclosures for saw pulleys

Fig. 4—Below—Vitreous-enameled stamped case utilizes existing production facilities to produce a neat, easily cleaned hand dryer for wall mounting



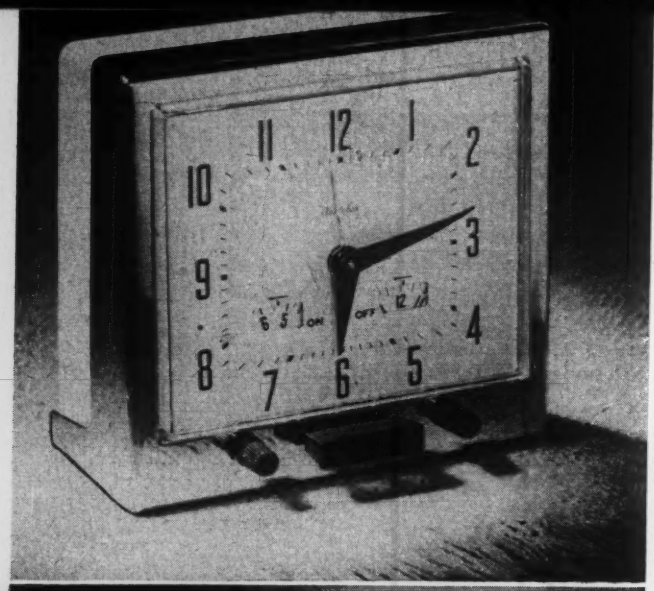


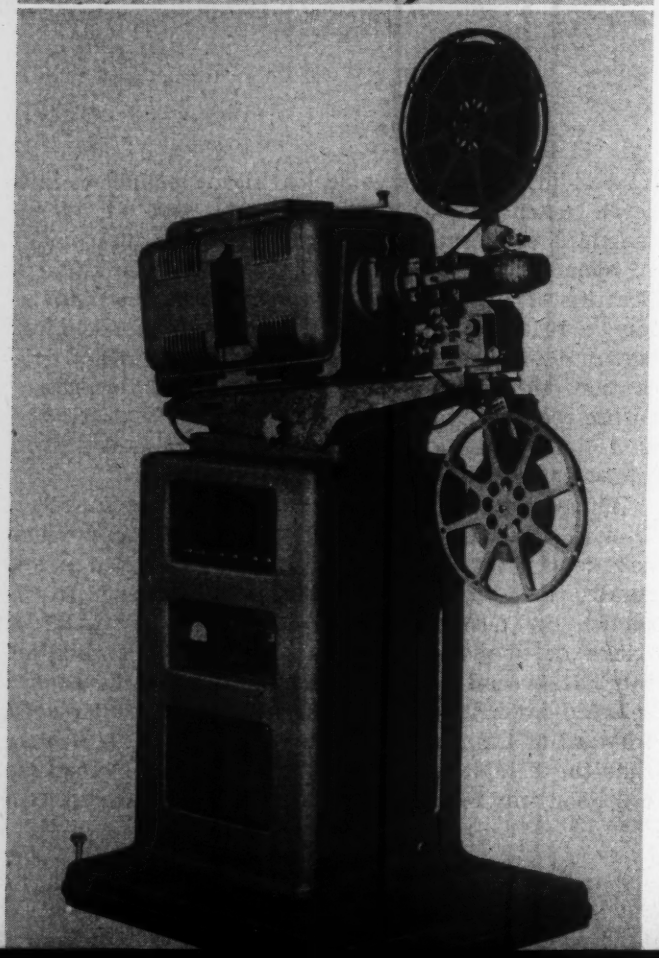
of design, without any tricks of speed lines or imitation engine-turned sheet-metal panels, expresses the reputability of the manufacturer. Sand-cast aluminum frames eliminate the necessity for a sheet-metal housing. The right-hand casting carries bosses for mounting pulleys and adjustment levers, and the left-hand casting serves as an access door and protective cover. The taper from top to base keeps the overall size at a minimum but provides base width to establish stability in actuality, as well as in appearance.

The vitreous-enameled stamped case for the National Dryers Corp. hand dryer, *Fig.* 4, exemplifies the same general thinking applied to a drawn shape. The simple contours are an obvious requirement for this type of equipment because ease of cleaning is one of the greatest factors to be considered. Drawn metal with the vitreous-enamel finish was second in desirability to stainless steel, but economics and existing plant equipment made the selected material and finish more desirable from the production viewpoint.

Westclox electric timer clock, *Fig.* 5, relies on contrast of materials and color. A highly polished brass bezel, extended to form the base, and an ivory-enameled back shell combine to make the instrument at home in either the living room or the kitchen. Because of the requirements of the time-set indicators, a rectangular dial for the clock provided better layout than a round dial. Devoid of unnecessary decoration, the dial has large numerals to permit accurate reading from a distance. The dial background is a light sliver-gray satin finish with dark brown numerals to match the brown acrylic knobs.

Wittie Mfg. Co. Inc. makes an oscillating fan, *Fig.* 6, with heating element for year round use. The base is die cast and the fan housing is formed sheet. Again, the functional elements have been used to establish the design without resorting to meaningless tricks of shape or applied decoration. Method of fabrication had considerable influence on the design selected for manufacture. The heating element placed forward of the fan requires a concentrated air flow so that the heat is not distributed too widely. Because the hood was to be fabricated by simple forming, two





truncated cones proved to be the most direct design as well as the most pleasing. The base serves additionally as a means of tilt adjustment, a pivot for the oscillating action and an enclosure for the switch. Forms to complement the hood shape were developed for the base casting and two shades of metallic brown paint are used as finish.

Shown in *Fig.* 7 is a complete redesign of the Bell & Howell Filmo-Arc projector housing and pedestal which carries the rectifier and audio units. Requirements for heat dissipation and light shielding posed a complex set of requirements which were best answered by use of aluminum castings. The housing is a fawn-gray wrinkle finish with bright metal accents on the venting grids, control knobs, and working parts of the projector head. A rub rail on the base is polished aluminum. The control panels are black with a polished design element in the form of a rectangle used to achieve symmetry in an assembly of purchased components such as meters, knobs, etc. The bright surface, instead of the individual units, attracts the eye and gives a plan to a more or less unorganized area. The stamped sheet-metal louvre

Fig. 5—Left, above—Contrasting materials and color make a neat appearing timer clock

Fig. 6—Left—Oscillating heater fan has die cast base and simple housing of sheet metal

Fig. 7—Left, below—Redesigned housing and pedestal for projector show how miscellaneous parts may be grouped and treated to make an integrated unit

areas in the base are from stock dies. The access door on the back side and the entire top are also louvred. The pedestal base replaces a tubular frame on which the rectifier and audio units had been hung.

These products indicate that distinction and individuality can be attained in any material or by any manufacturing process without resorting to superficial devices. The purpose of styling has always been to create pride of ownership. Tricks and devices, many of them bad, have been used by industrial designers to capitalize on this basic urge.

As in all things, the honest endures. Styling today is more honest than it has been in the past but there is much room for improvement. There is still superficiality evident in consumer products, but there are fewer "speed lines," bulbous shapes, terraced forms, or "three well-placed lines" in evidence as there were ten years ago. To avoid use of these outmoded gimmicks it is necessary to rely on the judgment of those qualified through training and experience to establish pleasing co-ordination of forms, materials and color and to use these elements to achieve an end result in themselves.

To know how to oppose a curve with a line or to balance a mass with a point are tools of the profession; their use is successful in direct ratio to the skill of the designer. The trend in appearance design is toward greater honesty in establishing forms. Bulbous forms are out, the crisp approach is in.

Automatic Dynamic Balancers

Part 3—Designing for Complete Effectiveness

By Ernest L. Thearle
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Palo Alto, Calif.

IN THE first two parts of this series analyses were given of Leblanc-type (liquid) and mechanical-type self balancers. Although all these devices compensate for unbalance to some degree, it was shown that none fully satisfies the four basic requirements of the ideal self balancer:

1. Complete effectiveness, correcting for all unbalance
2. Effectiveness at low speeds to maintain balance as the critical speed is approached
3. Effectiveness at speeds above the critical to compensate for changes in unbalance due to extraction
4. Correction in plane of unbalance to minimize horizontal reaction at the elastic mounting.

Of most concern in the design of centrifugal extractors for domestic laundry machines and similar applications is the common failure of these earlier arrangements to correct for unbalance over the entire speed range. This concluding article presents a system, termed the Autobalancer, that is effective at all speeds.*

EFFECTIVENESS AND EFFICIENCY: Behavior of an elastically mounted rotating system, such as a centrifugal extractor, was described in Part 1, (Sept. 1950). *Fig.* 4 showed how the eccentricity, *e*, of the rotor lags behind the unbalanced weight by the angle of lag, ϕ. Starting at zero at zero speed, ϕ equals 90 degrees at the critical speed and approaches 180 at high speeds. It appears from this that any device which is to measure the eccentricity and deliver a proportionate amount of balancing fluid to a point opposite the unbalance must be sensitive to speed and thus correct for this changing angle of lag. At first glance it may be thought that, in order to meet this requirement, the device must deliver each increment of the balancing fluid to a point opposite the unbalance. However, this is not the case. In *Fig.* 14, *W* represents the initial unbalance in the system. Because this unbalance has both magnitude and direc-

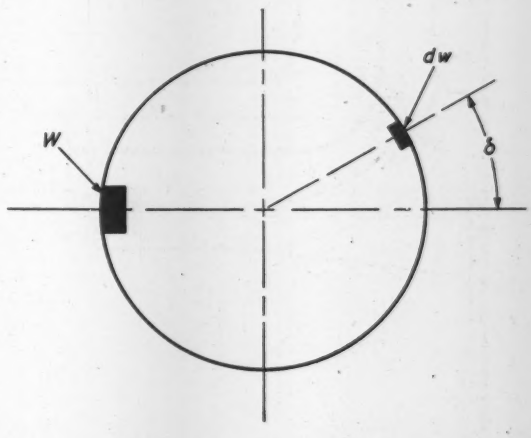
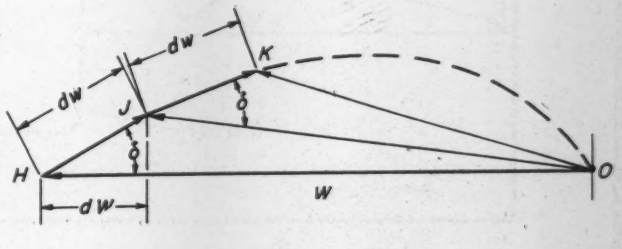


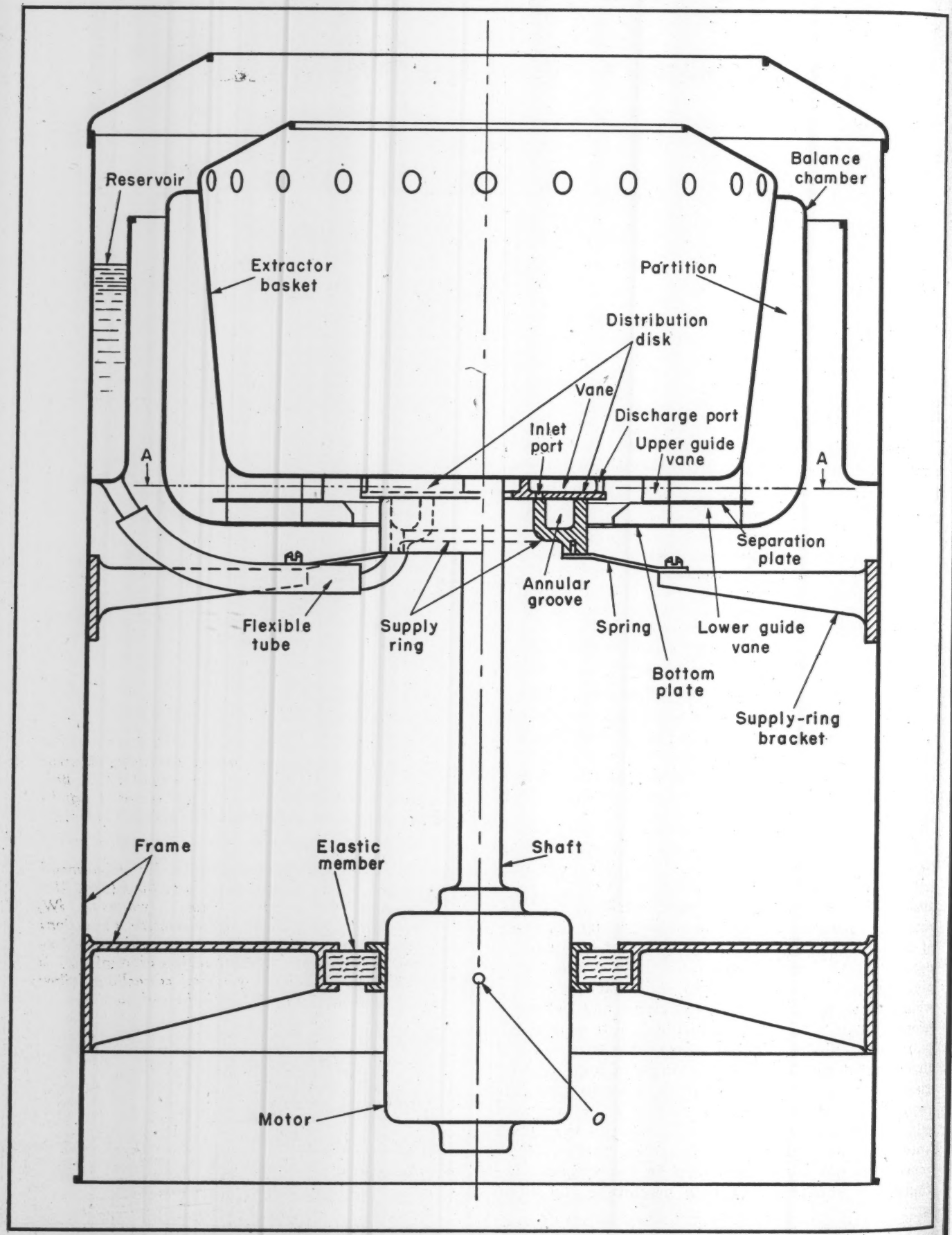
Fig. 14—Above—Diagram showing increment of corrective weight (dw) acting at angle of error (δ) with respect to unbalance weight (W). Balance is achieved because of the vectorial significance of successive increments, as shown in Fig. 15

Fig. 15—Below—Vector diagram for unbalanced weight (W) and incremental corrections (dw). Unbalance is compensated by sum of values of (dW) which are components of successive corrections (dw) acting at angle of error (δ) with each preceding resultant. If mean value of angle (δ) is less than 90 degrees, vectorial corrections follow spiral path to produce balanced condition



* Patent pending.

Fig. 16—Vertical cross section through centrifugal extractor employing Autobalancer system. View at Section A-A is shown in Fig. 17



tion, it may be represented by the vector *O-H* in *Fig.* 15. Suppose that the balancing device, during a small interval of time, adds the increment of corrective weight, *dw,* not directly opposite the unbalance but at an angle, δ, from this position. Let this angle, δ, be known as the *angle of error.* The increment of correction, *dw,* at the angle, δ, is shown in *Fig.* 14 and is represented by the vector *H-J* shown in the diagram of *Fig.* 15.

The next increment of correction, *J-K,* will then be added at an angle of error, δ, from the vector *O-J,* which is the resultant of the original unbalance *W* and the preceding increment of correction, *H-J.*

It is seen that a completely balanced condition will always be approached if the angle of error, δ, is less than 90 degrees. This condition is approached along a logarithmic spiral as shown in *Fig.* 15. Thus the *effectiveness* of such a device is 100 per cent as long as the mean angle of error, δ, is less than 90 degrees.

From *Fig.* 15

$$dW = dw \cos \delta \quad \ldots\ldots (44)$$

For a constant angle of error, or with δ taken in the sense of a mean angle of error, Equation 44 integrates to

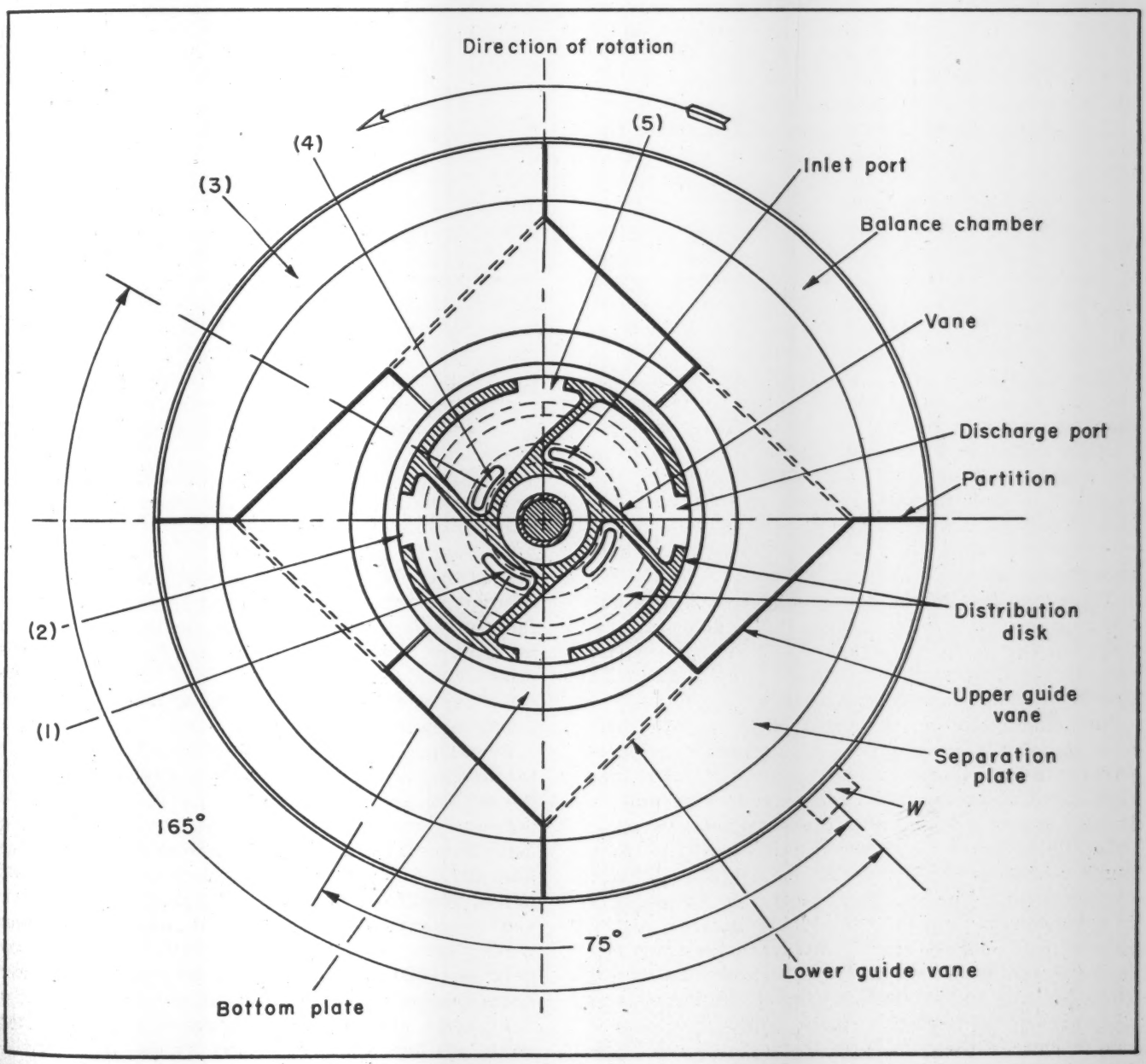
$$W = w \cos \delta \quad \ldots\ldots (45)$$

With the ratio, W/w, defined as the *efficiency* of the balancer, Equation 45 shows that

$$Efficiency = \cos \delta \quad \ldots\ldots (46)$$

Thus, for example, if the mean angle of error is 30 degrees and each increment of the correction is added at the same radius as the original unbalance, each pound of corrective fluid will balance 0.866-pound of

Fig. 17—Section A-A of extractor shown in vertical section in Fig. 16. Autobalancer offsets angle of lag by deflecting balancing fluid so that angle of error is never greater than 45 degrees



original unbalance. In other words, the efficiency is only 86.6 per cent because of the distribution of the corrective balance weight.

The fact that the efficiency of such a device is less than 100 per cent does not detract from its value; it is still 100 per cent effective in achieving balance.

In the design to be described here, an effort is made to reduce the angle of error to a minimum over the most important parts of the operating speed range.

PROPOSED DESIGN: The Autobalancer consists essentially of:

1. An elastically mounted rotor carrying on its outer periphery a number of compartments for receiving balancing fluid
2. A valve for introducing this fluid to the rotating system at a point determined by the eccentricity of the rotor
3. Speed-sensitive distributing means to correct for the angle of lag and deliver this fluid to the proper compartments.

In *Fig.* 16 is shown diagrammatically a vertical cross section through a typical centrifugal extractor upon which are mounted the elements of an Autobalancer. In *Fig.* 17 is shown a horizontal cross section through the extractor of *Fig.* 16 in the plane *A-A*.

The rotor or basket to be balanced, containing the unbalanced load, is mounted on the shaft which is, in turn, mounted on the motor. The motor is supported in the frame of the machine by an elastic member such as a rubber ring, rubber blocks or springs, so that the whole rotating system may oscillate about the point *O*. This elastic member is designed so that the natural frequency of oscillation, or critical speed of the system, is quite low—between 100 and 200 rpm.

Fixed to the bottom of the basket for rotation with it is a distribution disk. A supply ring, which does not rotate, is supported by flat springs attached to brackets on the frame of the machine. These springs, radially stiff in order to center the supply ring, press the supply ring upward against the bottom of the distribution disk to form a liquid seal between the two. The supply ring may be molded plastic or some other material that will lubricate properly with whatever fluid is used.

The upper face of the supply ring contains an annular groove which is supplied with fluid from a reservoir (or pump) by means of a flexible tube. The reservoir is filled with the extracted fluid during the first few revolutions of the basket.

Surrounding the basket and fixed to it is the balance chamber which is divided into a number of compartments by means of radial partitions. In the design shown there are four of these compartments, although any number greater than two may be used.

In the bottom face of the distribution disk are a number of slots or inlet ports, one for each balance compartment. The outside diameter of these ports is approximately equal to the inside diameter of the groove in the supply ring. Thus when the eccentricity of the rotor is zero, all these ports remain closed. However, any radial displacement of the basket and distribution disk, relative to the supply ring, opens one or more of the inlet ports on the "high side" (direction of eccentricity) and permits fluid to enter the distribution disk at these points.

Due to centrifugal action, this fluid flows outward on the distribution disk through the channels formed by vanes and is expelled through corresponding discharge ports. As shown, the discharge ports lag behind their corresponding inlet ports.

Fluid thus discharged from the distribution disk enters its appropriate balance chamber compartment by either one of two sets of channels, depending upon the rotor speed. These channels are separated by a horizontal plate and formed by upper and lower guide vanes.

At low rotative speeds, below the critical speed, the velocity of the fluid passing through the discharge ports is low. This fluid falls on the bottom plate and enters the balance chamber through one of the lower channels. The balance chamber into which the fluid is directed by the lower guide vanes is displaced from the eccentricity by a relatively large angle in the counterrotative direction. Corrective fluid is thus placed in a region generally opposite the unbalance, compensating for the low angle of lag at low speeds.

The inner diameter of the bottom plate may be so proportioned that the balancer becomes effective only above some prechosen rotative speed.

Correction Angle Less at High Speed

As the speed of the rotor is increased, the velocity of liquid discharged from the ports increases until, at some prechosen transition speed, the fluid falls onto the separation plate and enters the appropriate balance compartment through the upper channels. As shown in *Fig.* 17, the balance chamber receiving fluid from the upper channels is displaced by a relatively small angle in the counterrotative direction. For these higher speeds and thus greater lag of the eccentricity, less angular compensation is needed to place the corrective fluid in an effective balancing position.

The inner diameter of the separation plate may be so proportioned as to give the transition speed any desired value.

The particular design shown in *Fig.* 17 is so arranged that the angle of error is zero at two different rotative speeds: one just below the critical where the angle of lag is 75 degrees, and the other at a high speed where this angle is 165 degrees. This choice gives the device a maximum efficiency as the rotor approaches the critical speed where it is most sensitive to unbalance.

For example, let the unbalance be represented by the weight, *W, Fig.* 17. At some low speed, just below the critical, the angle of lag of the eccentricity is 75 degrees. Fluid is then admitted to the distribution disk through port (1) and discharged through the port (2). Because the discharge velocity of this fluid is low, it will fall on the bottom plate and enter balance compartment (3) as determined by the lower guide vanes shown by dotted lines. Balance compartment (3) is directly opposite the unbalance; hence the angle of error is here zero.

At some high speed above the critical, when the angle of lag of the eccentricity is 165 degrees,

fluid enters the distribution disk through port (4) and is discharged through port (5). Because the discharge velocity of the fluid is high, it falls on the separation plate, is guided by the upper guide vanes shown by the solid lines, and enters the same balance compartment (3), which is directly opposite the unbalance. Thus the angle of error is again zero at this high speed. If the inner diameter of the separation plate is designed to give the proper transition speed, the angle of error is always less than 45 degrees. Hence, the effectiveness of this device is 100 per cent and its efficiency greater than 70 per cent.

CONCLUSION: In the particular design described, four balance compartments are used. Obviously the device need not be limited to this number inasmuch as any number greater than two would be operative.

The design shown here permits the designer to choose arbitrarily:

1. A minimum rotative speed at which the balancer becomes operative
2. A low speed range over which the angle of error is less than 45 degrees
3. A high speed range, above the critical speed, over which the angle of error is always less than 45 degrees
4. The transition speed between these two ranges.

It is of interest to note that during the balancing operation fluid is entering the spinning member from rest. Angular momentum of the rotating system is therefore increasing due to the added fluid. This tends to hold the speed down until the balancing operation is completed and thus avoids passing through the critical speed before the system can do so smoothly.

Added Stop Checks Eccentricity and Speed

A method of obtaining the same effect in a more positive manner, at the same time saving space, is to provide a definite stop to prevent the eccentricity from becoming any greater than that necessary to open the inlet ports. Then, as the rotor approaches the critical speed, the shaft contacts the stop and the resulting friction prevents an increase of speed until the balancing operation is almost completed. With the shaft in contact with the stop, the angle of lag of the eccentricity is slightly less than 90 degrees, the condition for maximum efficiency of the balancer. When the balancing operation is about completed, the shaft leaves the stop and the speed increases. This accomplishes most of the balancing operation at the speed at which the rotating system is most sensitive to unbalance, and thus assures a good balance at the higher speeds.

Where it may be difficult to level the machine and thus properly center the supply ring, the ring may be made self centering by mounting it to slide radially against some friction. This provides some desirable damping for large eccentricities, which disappears after the balancing is completed.

Where the extraction process is continuous, the operation of the balancer may be made continuous by providing the proper size of leaks in the balance compartments with less overlap between the supply ring and the inlet ports.

It should be noted that with the form of valve described, having a nonrotating member mating with a rotating member, friction is not confined within the rotating system and therefore causes no whipping at speeds above the critical.

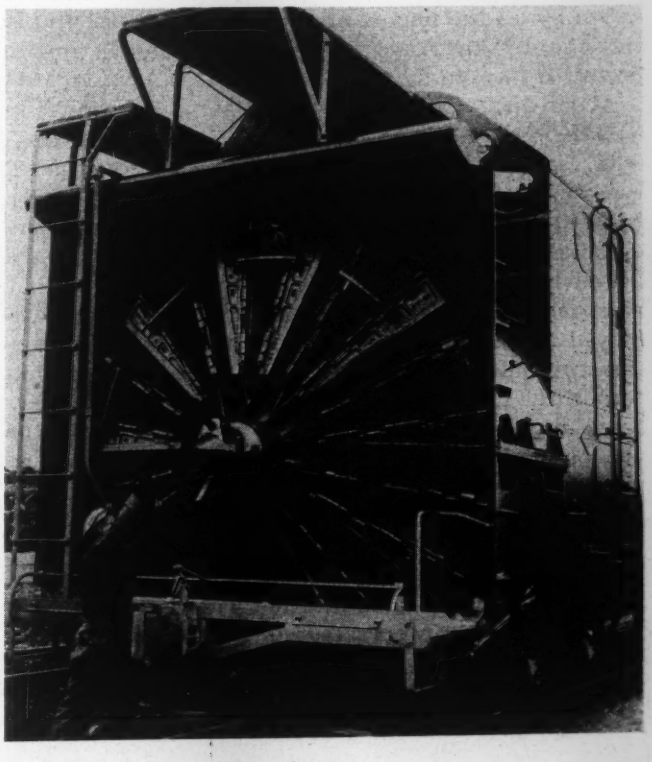
The Autobalancer described has the following characteristics:

1. It corrects for all of the unbalance
2. It is effective at speeds below the critical and operates to balance the rotor as it approaches and passes through the critical speed
3. It is also effective at speeds above the critical, correcting for changes in unbalance caused by extraction or other disturbances at these high speeds
4. It adds the corrective weight in the plane of the unbalance, thus minimizing the horizontal reaction in the plane of the elastic mounting.

Thus, it is seen that the Autobalancer meets the requirements of the ideal automatic balancer.

Plow Burns Diesel Fuel

FORMIDABLE-LOOKING business end of a 12-ft rotary snow plow built for the Union Pacific is shown in the accompanying illustration. Because of the general availability of diesel oil as a fuel, the steam-powered plow has been equipped with a diesel oil burner having a rating of 1500 hp. Two separate 3-cylinder steam engines furnish power to the blade, with overspeed governors on the engines to limit rotor speed to 150 rpm. These governors prevent running away of the rotor when the plow breaks through a drift, a disadvantage of previous models. The 297,000-lb units are built by Lima-Hamilton Corp.



MILESTONES IN DESIGN

By John Kremer
Cleveland, Ohio

The Typewriter

THOUGH the first typewriters permitting a person to write faster than with a pen were not built till the 1850's, one of the essential components was introduced at about the same time printing from movable type was invented in 1450. This was the keyboard, the first contrivance that would allow touch operation of a complex percussive mechanism by all the fingers of both hands, applied to the clavichord in the fifteenth century and later in a more perfected form to the piano.

About 1850, Sir Charles Wheatstone, who already had the dial telegraph invention to his credit in England, and Dr. Francis, of New York, constructed typewriting machines with the piano as a basis. Both realized that the piano had reached a state of mechanical development where only the skill of the player limited the rapidity of striking notes, and they reasoned that this same construction might be employed to allow all the fingers of both hands to function conveniently and quickly to imprint selected letters in the desired order. It remained to provide lettered keys arranged to impinge at a single point on the paper, and to move the paper or the printing mechanism after each letter had been struck.

Francis' typewriter, *Fig.* 1, was modeled closely on the action of the piano and allowed writing faster than by pen, but as it was delicate and of cumbersome construction, simplification was called for. How this was done in a successor to the Franics machine is shown in *Fig.* 2. This is a front view of a schematic form of the Densmore typewriter, which Thomas A. Edison had a part in perfecting, showing the principles of the printing action.

Because this machine was produced in America before the general application of the principle of interchangeable parts, it was constructed so that fitting and adjustments were easily accomplished with wire take-ups. A single set of pivots had to be accurately aligned to provide for even typing. This Densmore machine, though far from perfect since the keys did not always respond readily, was the first to have wide sale and to prove that a large market existed for typewriters.

In order to minimize the clashing together and sticking of type bars, Densmore had studies made of the relative frequency of occurrence of juxtaposition of various letters of the alphabet in English text and then arranged letters on the keys so that there would be the least tendency for the bars to strike against each other.

Origin of the Standard Keyboard

Mark Twain, one of the earliest typewriter purchasers, when asked for a testimonial said that he had entirely stopped using it. He declared it was ruining his morals, that it made him want to swear. But he was the first author to submit a typewritten manuscript to a publisher.

Though the visible writing feature has almost no importance for the practiced touch-typist, it was a most significant development when introduced. Many tried to achieve it and partially succeeded by means such as the radical expedient of running the ribbon

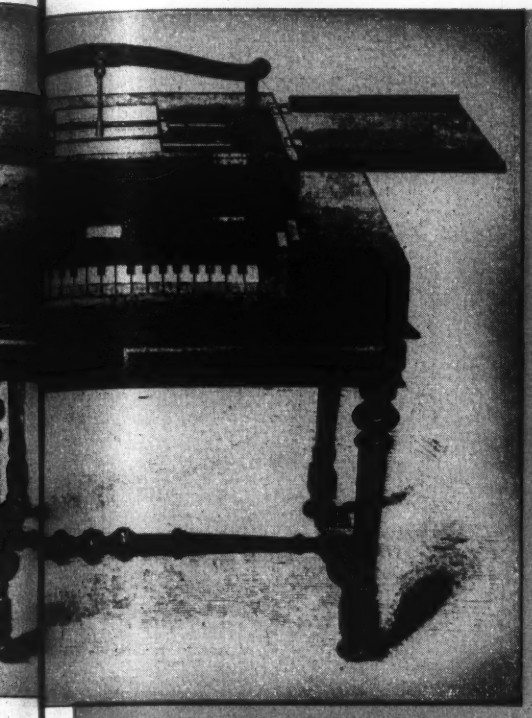


Fig. 1—Above—Francis' typewriting machine, 1857

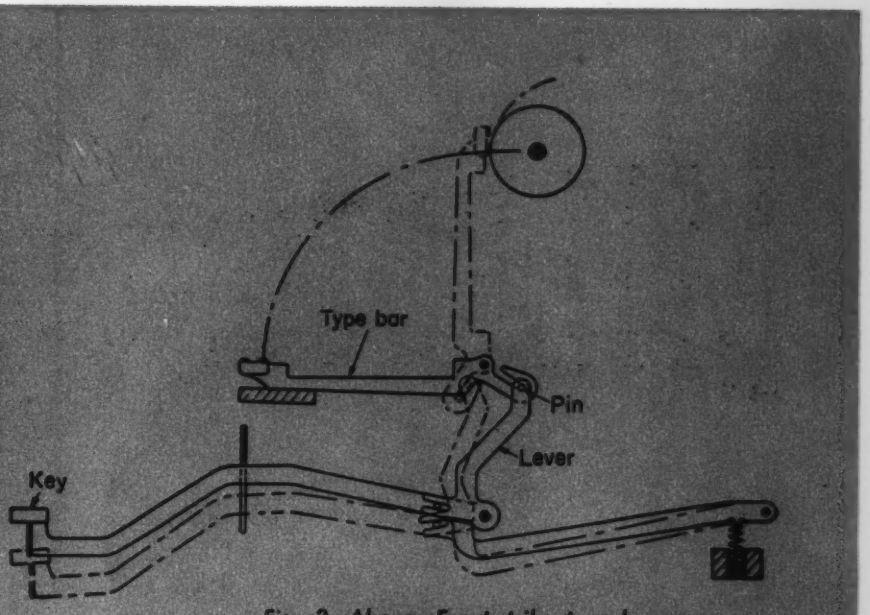


Fig. 3—Above—Front-strike type-bar mechanism of late 19th century

Fig. 2—Below—Densmore typewriter mechanism

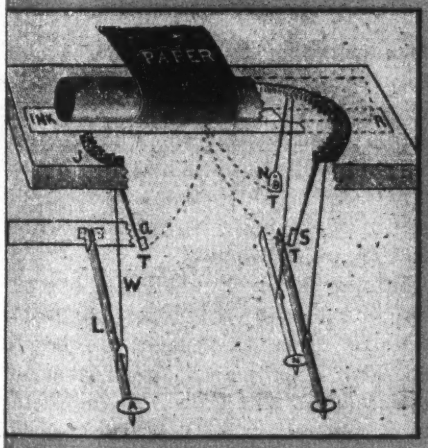
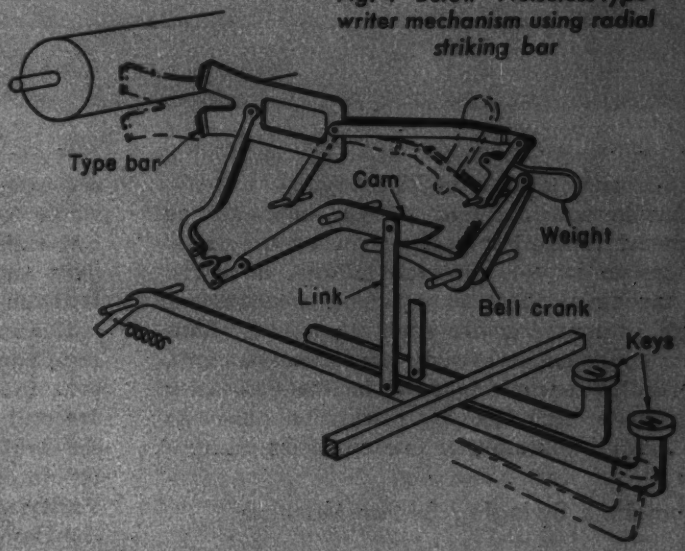


Fig. 4—Below—Noiseless typewriter mechanism using radial striking bar



at right angles to the lines of writing. The type-bar mechanism illustrated in *Fig.* 3, first used in 1897, was probably the simplest and the most successful ever devised and provided a clear view of the typed line. Depression of the key causes the lever and the pin engaging with a slot in the type bar to swing the type bar upward to the horizontal center line of the platen. Change of case by rise of the platen, by a single shift mechanism, is not shown.

When large offices were equipped with many typewriters, late in the 19th Century, the noise they made as the keys hammered against the paper became disagreeable. To relieve this a pressure-actuated printing mechanism called the "noiseless typewriter" was introduced, *Fig.* 4. In modern "noiseless" designs, each type bar is operated by either of two keys, giving two characters. On deprsession of the key, the link engages a cam and bell-crank. The latter moves the type bar close to the platen, over-throw of the "momentum accumulator" weight making a pressure contact. A linkage (shown for one key only) determines which character comes to the platen.

This noiseless construction followed many years of unsuccessful effort by the Colby's, father and son, until, so the story goes, an engineer of theirs, Nils Anderson, got the idea of the mechanism just de-

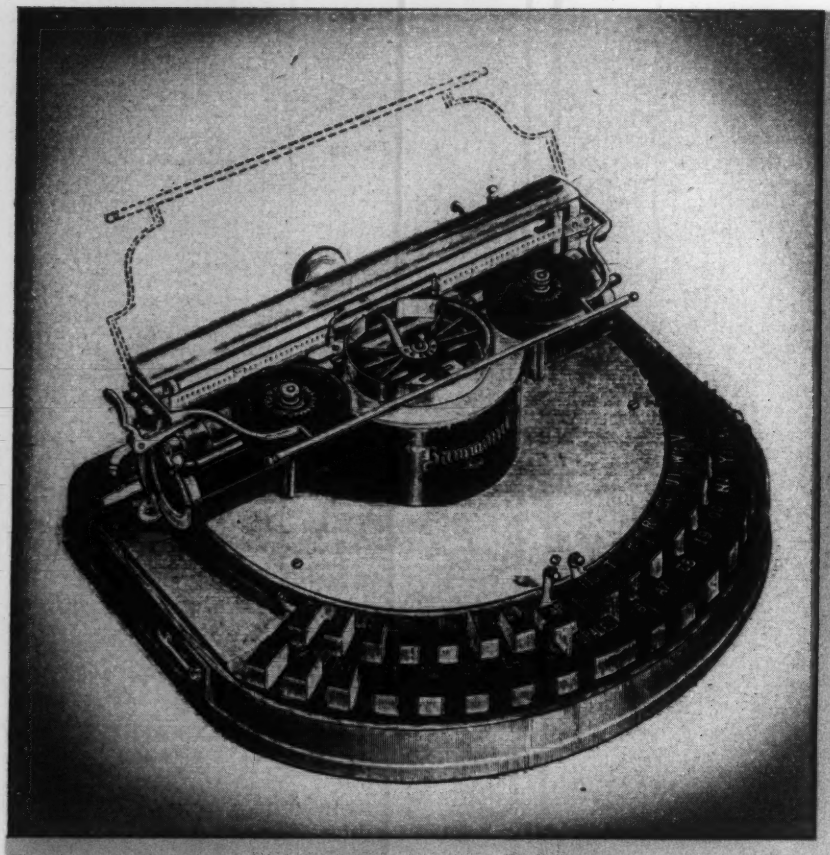
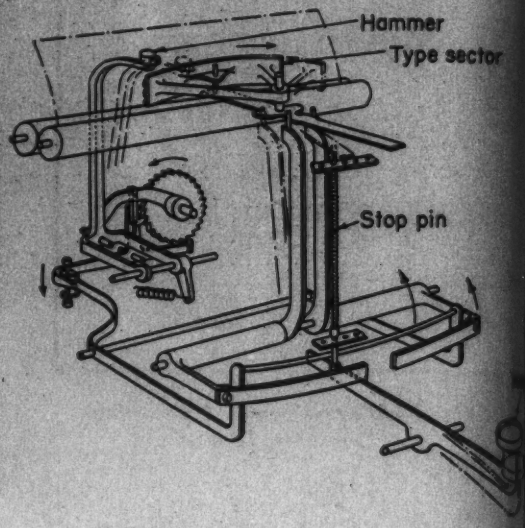


Fig. 5—Left—Hammond typewriter, developed about 1880

Fig. 6—Below—Swinging sector mechanism of Hammond typewriter



scribed, while sitting in a hotel lobby.

A machine in which a font of type is moved as a whole relative to the impression point on the paper would appear to have several advantages over the type bar machines. The bars, being nonexistent, cannot possibly interfere with each other, thus eliminating a major evil in the early nonservo machines. More important, the whole font can be changed easily to an entirely different one by merely slipping one font off a spindle and another one on in its place.

Hammond's Swinging Sector Machine

With these ideas in mind, and to reduce the mechanical contrariness that made Mark Twain want to swear, James B. Hammond, in the period ending 1880, developed and patented a revolutionary typewriting machine, *Fig.* 5. Hammond was known as an expert at shorthand, a reporter and editor and as a translator of the classics. As a relief from the rigors of scholarship, Hammond undertook to devise a typewriter with none of the faults of the current type-bar machines and turned to the swinging sector style which was then, as now, in eclipse.

His great contribution, aside from improvements in keyboards, visibility and accuracy, was to supply the mechanical energy necessary for hammer impression by means of a spring. The spring was tensioned by action of the arm of the typist as the carriage was returned, rather than by the finger striking each key. Thus, Hammond's was the first servomechanism typewriter. Even though it had a whole font of letters to be moved by finger muscles for each character typed, the touch could be kept light and the speed of typing high since the energy for type impression was spring supplied.

Hammond machines, distinguished for good construction and smooth operation, were at one time sold in considerable numbers and their successors are still used for making the originals for photo offset work. They are also used with interchangeable typefaces and to justify lines of type as in today's Varitype machines. *Fig.* 6 shows the mechanism of the Hammond. Depression of the key raises a selector stop pin and, by movement of a bell-crank lever, rotates the type sector until a lever attached to the sector meets stop pin. A ratchet mechanism then releases a hammer which strikes the paper against the sector.

After the type bar and type-sector machines were developed, tens of millions of the former were sold because operators found they could write more legibly and rapidly, but the machines were not portable. A demand for a machine of light weight that could be easily carried around was thus created. One of the first of the portable machines, made partly of aluminum and weighing less than a dozen pounds, was the Blickensderfer of about 1896, shown in *Fig.* 7.

The Blickensderfer mechanism may be seen in *Fig.* 8 where half of the keys operate one of the gear quadrants and the other keys operate the other gear quadrant. Depression of the key first engages the

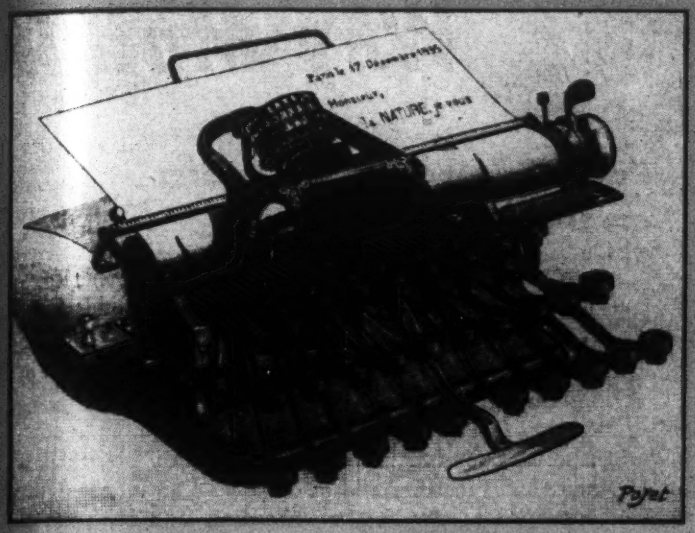


Fig. 7 — Above — Blickensderfer typewriter, popular in Europe about 1896

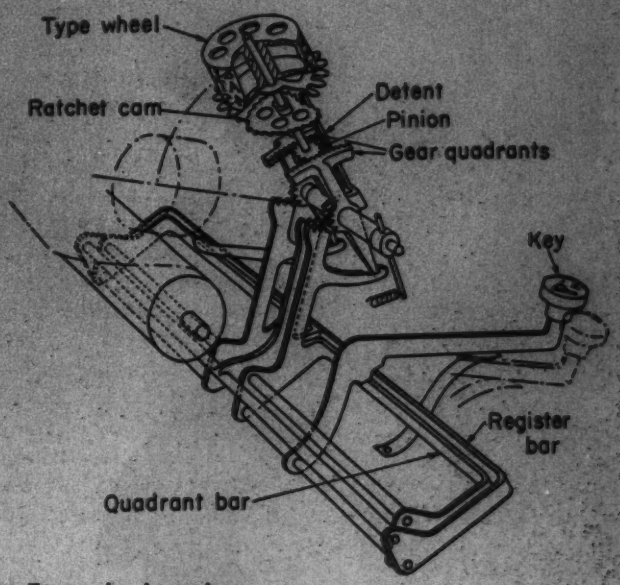


Fig. 8—Right—Type-wheel mechanism of Blickensderfer machine

quadrant bar; the gear quadrant rolls the pinion and the type wheel to which it is attached downward until the key engages a register bar, and a detent engages the appropriate tooth of the ratchet cam, when the whole assembly moves downward to the platen. The mechanism for change of face by axial double shift of the type wheel is omitted. As the fingers of the typist operating the Blickensderfer had to rotate and translate the type sleeve a considerable distance to print some letters of the alphabet, the touch was uneven and rather heavy, and the speed was slow. Application of an electric motor to drive this typewriter made it one of the first electric machines.

Ingenious Typing Systems

Later, to obtain the advantage of light touch, rapid action and enough impressing power to make twenty carbon copies, type-bar machines were power driven by air, solenoids and friction clutches of the type applied to punch presses. Record speeds of 180 average five-letter words a minute, rivaling the speed of dictating machines, have been attained on them.

All the foregoing machines have contributed features to designs being manufactured today. But there are three others that, although not represented here, were certainly ingenious and may some day be put to general use. One of the simplest and cleverest methods of controlling the impressing of type on a platen was employed in the Soblik machine. Keys with pin holes in their centers were immovably fixed in position at the front of the machine. When the fingers touched the keys they blocked the flow of air through these holes, changing the pressure within associated conduits kept under pressure. This caused a hammer to strike the paper against the selected type letter embossed on a sleeve kept in continuous rapid rotation.

In the multiple key-bank DeCarsdale and Schiesari machines, a whole syllable was typed in the time usually required for a single letter. Something like 70 words a minute were claimed to have been typed on these machines, though the mechanism was crude and nonservo.

Flowers, about 1912, by using a system of tuned reeds and associated resonant electric circuits, made a typewriter capable of automatically typing spoken vowel sounds; perhaps some day these principles and the greater knowledge of sound, electricity and speech may make a voice-operated typewriter a reality.

"Atomic energy as a source of power for aircraft promises airplanes combining extremely high speed with almost unlimited range. Atomic power offers fuels such as uranium that would not be consumed nearly so fast as those in chemically fueled planes. High operating temperatures, heat transfer, and weight of the mass needed to stop radiations are some of the problems encountered"—ANDREW KALITINSKY, *chief engineer of NEPA project, Div. of Fairchild Engine and Airplane Corp.*

Why Machin

Part 7—Failures Originating on the Drawing Board

FRACTURE statistics indicate that in a number of industries fully 30 per cent of all fatigue failures can be traced directly to faulty design. Inasmuch as these failures originate on the drawing board, it is obvious that the time and effort spent in emphasizing the concepts of stress concentration and its alleviation should be well rewarded. The number of service failures resulting from improperly designed fillets, keyways, oil holes, and other geometrical discontinuities is large. Such a fracture is expensive, not only in the time and material invested in the fractured piece itself, but also in probable damage to the mating pieces and surrounding machinery, injury to the operating personnel, and risk of a time-losing shutdown if the member happens to be part of a production machine.

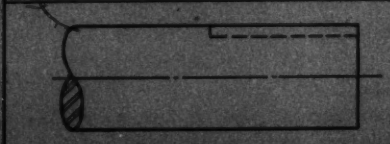
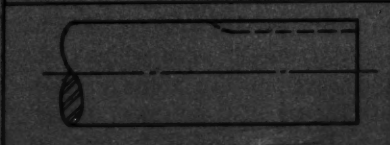
KEYWAYS: Improperly designed keyways represent a surprisingly large portion of design failures. Peterson has demonstrated that end-milled keyways can reduce the fatigue strength of medium carbon steels by 40 per cent and of chromium nickel steels by over 50 per cent[1]. Face-milled sled-runner type

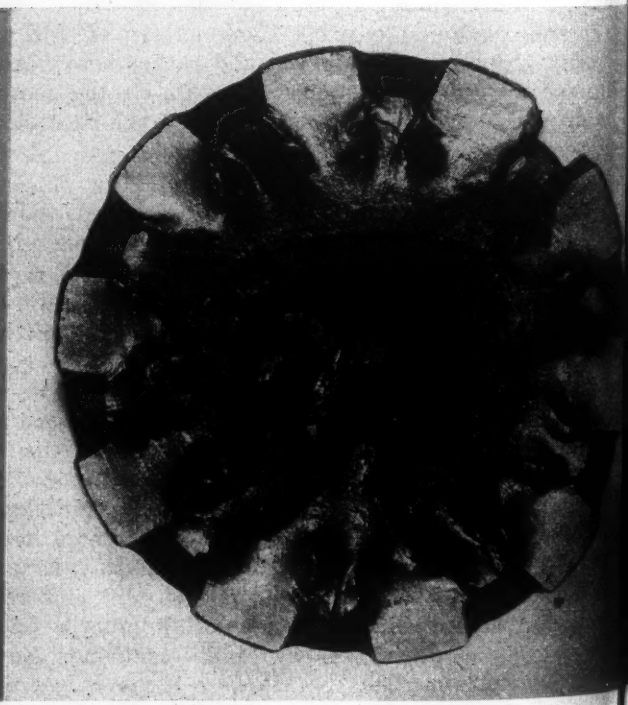
[1] References are tabulated at end of article.

Photo, courtesy Briggs Manufacturing Co.

Fig. 61—Above—Position of stop marks indicates fracture originated at corner of keyway in this fatigue failure of a large shaft

Table 4—Stress-Concentration Factors due to Keyways in Heat-Treated Steel Shafts

Profile Keyway	Bending	Torsion
	2.0	1.6
Sled-Runner Keyway		
	1.6	1.6



arts Fail

By Charles Lipson
Consultant
Detroit, Mich.

keyways are slightly less damaging to fatigue strength. The harmful effect of improper keyways is further emphasized by the data in TABLE 4[2]. Keyway fractures often start near one end of the keyway at the junction of the straight portion and the rounded tip. Stress amplification at this point apparently is due to the combination of the sharp radius in the bottom end of the keyway and the localized pressure from the corner of the key. This condition can be improved by designing the keyway with a more generous fillet at the bottom. It has also been found that as much as 20 per cent increase in fatigue strength can be gained by cold rolling the corners of the keyway.

In Fig. 61 is shown a fractured shaft from a large press. Examination of the fracture surfaces shows that the fatigue crack originated at the extremely sharp corner in the bottom of the keyway. Fractures of this type are very common in keyed shafts.

SPLINES: Splines are simply multiple keyways; therefore, generalities expressed for keyways hold true for splines. Fatigue fractures of splined shafts, however, show a very characteristic pattern. Because each spline represents approximately the same degree of stress concentration, fatigue cracks ordinarily originate almost simultaneously at the roots of all splines. These cracks then travel toward the center of the shaft at about the same rate, producing an appearance such as that illustrated in *Fig.* 62. This type of failure is essentially transverse with the center portion varying from flat to conical in form.

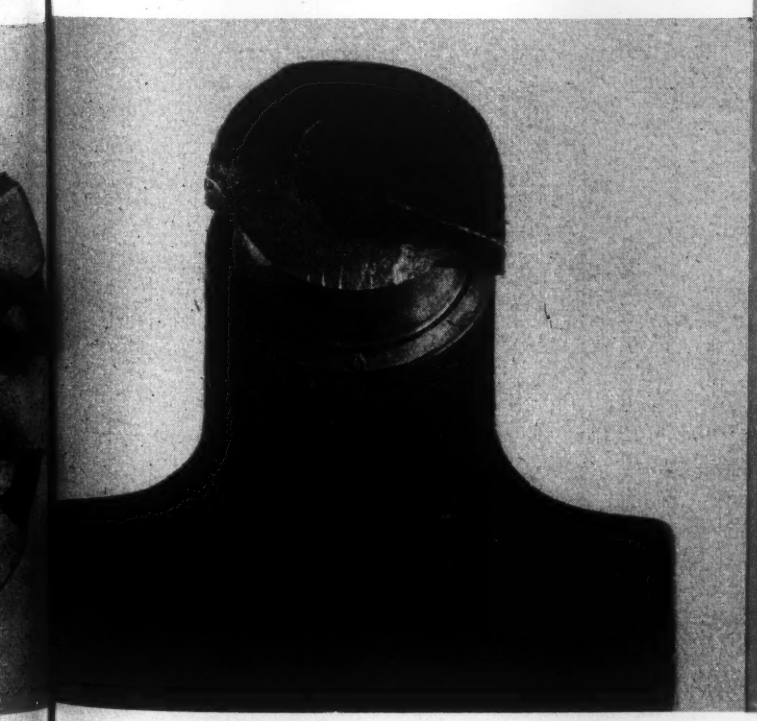
FILLETS: An inexcusable number of fatigue fractures are caused by inadequate fillets. The literature abounds with examples of failures arising from this cause. A great deal of research work has been done on the effect of fillets of varying degrees of severity, so that it is now possible either to design a fillet for the minimum practical stress concentration or to know quite accurately the degree of stress concentration to be expected if other design conditions limit fillet dimensions. Fatigue stress-concentration factors for various fillets[2] are given in TABLE 5.

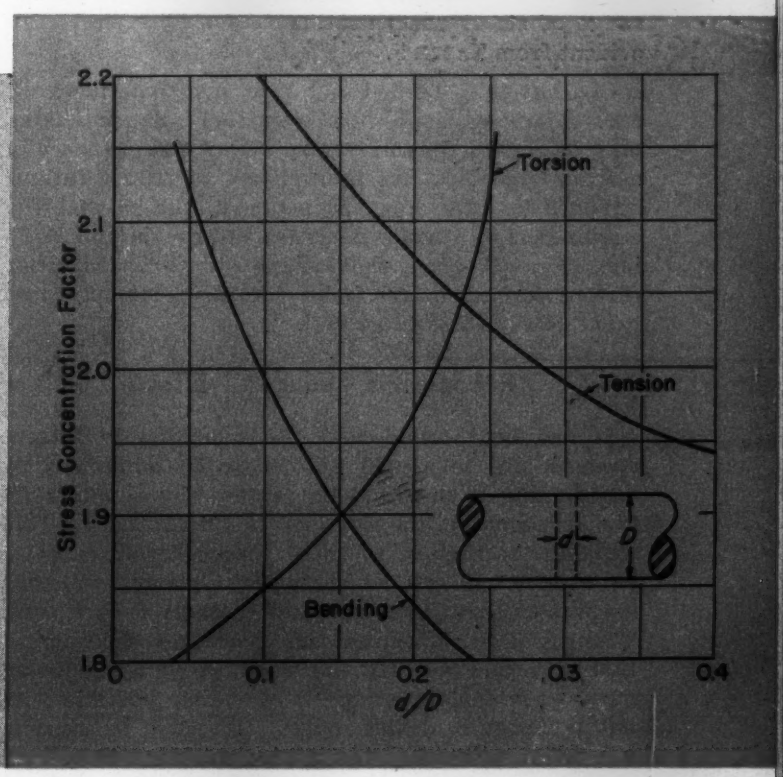
The fractured crankshaft, illustrated in *Fig.* 63, is a typical failure that originated in a small fillet radius. A larger fillet, consistent with the requirements for a sufficient journal bearing length, of course, would increase the fatigue strength of this part considerably. Oldberg and Lipson point out, however, that

Fig. 62—Far left—In this fatigue failure of a splined shaft a crack originated at the root of each spline, indicating high stress concentration. High overload also is denoted by the large instantaneous fracture zone

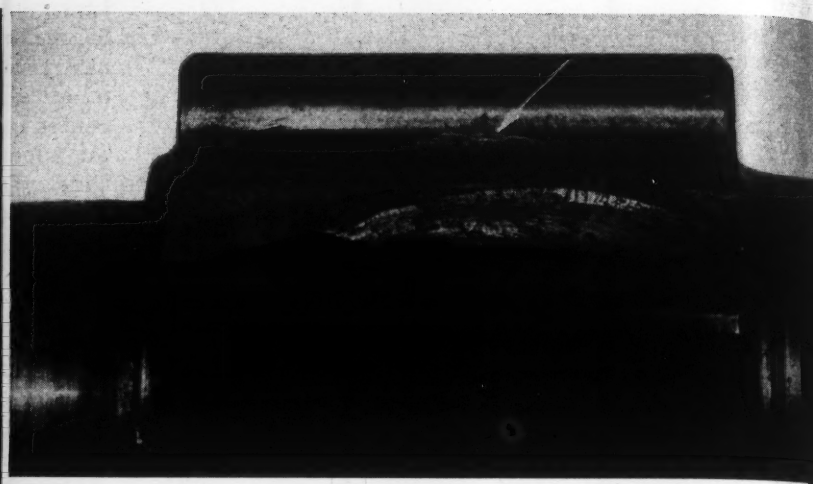
Fig. 63—Left—This crankshaft failed in fatigue because of stress concentration at the fillet. Beach marks indicate multiple points of origin of the fracture

Fig. 64—Below—This graph by Lipson, Noll and Clock gives stress-concentration factors due to a transverse hole in a heat-treated steel shaft





Photo, courtesy Briggs Manufacturing Co.



Photo, courtesy International Harv

Fig. 65—Left—Tension-fatigue failure of this tierod was due to stress concentration at the last thread and a change in section

Fig. 66—Above—In this fatigue failure of a gear tooth, the origin of the crack and its direction of propagation are clearly shown. Arrow points to another crack revealed by magnetic particle inspection of this 2.5-pitch pinion

Fig. 67—Below—This graph gives stress-concentration factors due to a press fit in a heat-treated steel shaft subjected to a bending load

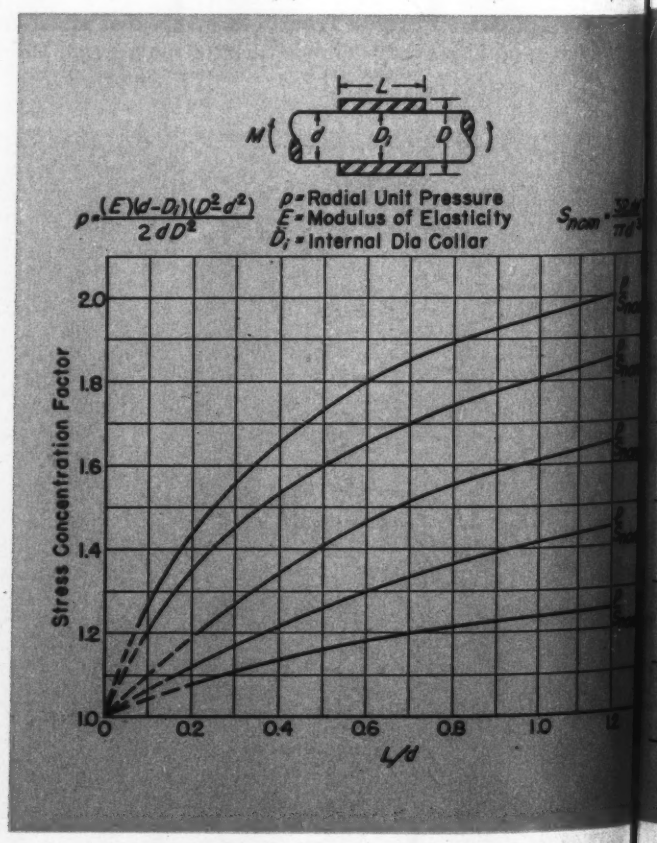
an increase in fillet size based purely on static stress values may be misleading[3]. Highly notch-sensitive steels benefit much more from an increase in fillet size than materials of low-notch sensitivity, such as cast iron. For instance, at an r/d ratio of 0.02 a 50 per cent increase in fillet radius decreases the stress-concentration factor by 15 per cent for 4340 steel but only 2 per cent for cast iron. It is also worth noting that the relative decrease in stress-concentration factor for a given increase in fillet size is not constant. Percentage improvement varies over the range of stress concentration versus fillet size, as is evident from TABLE 5.

OIL HOLES: Drilled holes in machine members, such as transverse oil holes commonly used in shafts, may be regarded as necessary evils. The presence of a transverse hole has been shown to reduce fatigue strength by more than 50 per cent, the effect being amplified if its surface is scored and rough from drilling. *Fig.* 64 gives the fatigue stress-concentration factors for transverse holes for the commonly used range of dimensional ratios[2].

Refinements Reduce Ill Effects

It has been found that the reduction in fatigue strength due to a drilled hole can be minimized by maintaining a smooth hole surface, by countersinking or rounding the edge of the hole and by cold working the edge of the hole. Some benefit can also be expected in critical cases from placing relieving grooves adjacent to the edge of the hole at both ends. These grooves, called *unloading notches,* in combination with cold working can bring about a significant increase in fatigue strength over that possible with an unimproved drilled hole[4]. It must be remembered, as with all other geometrical stress raisers, that the relative notch sensitivity of the material fixes the final significance of stress concentration.

THREADS: A single, circumferential, sharp V-notch represents a severe degree of stress concentration. Fortunately, fatigue stress concentration for most threads is not as great, the thread being seldom formed to the theoretical profile at the root because



of tool wear. In addition, stress fields created by the individual V-notches of a given thread interfere with each other, mitigating the otherwise severe stress concentration.

Failure Most Likely at Last Thread

Most bolts fail in fatigue at or near the last thread because this thread does not receive full benefit of stress-field interference from other threads. Hence, it is considered good design practice to reduce the body diameter of the bolt to slightly less than the root diameter of the thread. In bolt and nut combinations, failure often occurs at the plane where the bolt enters the nut because the greatest portion of the load is transmitted from the bolt to the nut via the first few threads of the nut. The effect is similar to that of an unfilleted change in section and results in stress concentration at this point. This case is illustrated in *Fig.* 65 which shows a fatigue fracture of a large hydraulic press tie rod. Failure occurred after 175,000 cycles of stress on the plane of the first thread in the female member. Although fatigue stress-concentration factors of approximately 4.0 have been observed in highly notch-sensitive steels[5], the smaller values given in TABLE 6 can be used safely in most cases as design guides[2].

Improvement of bolt strength can be gained by cold working the thread roots or by using threads that are roll-formed instead of machined. British Whitworth threads are also slightly superior to the old American National Standard threads because of their less severe profile. Considerable benefit in fatigue strength can also be obtained by redesigning the associated nut for better load distribution, such as by employing some form of re-entrant fillet on the loaded side of the nut. In bolt or stud failure, the designer should also look for nonaxial loading which can cause bending stresses greatly in excess of nominal stresses due to tension only.

GEAR TEETH: Because of the number of variables involving tooth profiles, helix angles and other design parameters, there are few data available on stress-concentration factors for gear teeth other than plain spur gears[6]. As might be expected, however, a gear tooth usually fails from a fatigue crack originating at the tooth root, assuming that excessive surface wear does not occur first. Nominal stress concentration at the root portion of the tooth profile may be aggravated by rough machining.

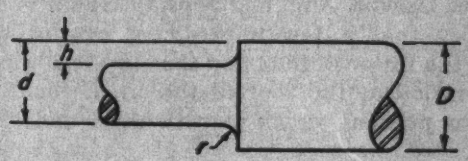
A typical gear-tooth fatigue failure is illustrated in *Fig.* 66, a photograph of a 2.5-pitch pinion with one tooth completely broken off and a second, indicated by arrow, showing signs of incipient failure.

Gear teeth are also subject to failure by fatigue pitting, but this form of failure is largely a function of surface contact stress.

PRESS AND SHRINK FITS: Although the harmful effect of a sharp inside corner is rather universally appreciated, designers sometimes fail to realize that a similar condition exists when a shaft has a collar clamped to it, or when a press fit assembly is made without any planned distribution of local stresses.

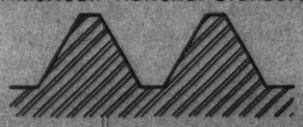
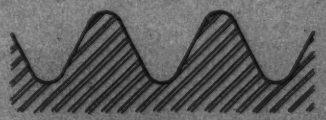
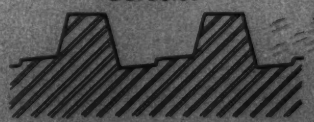
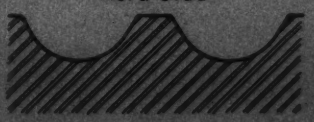
It has been shown experimentally that the endurance limit of a smooth shaft dropped from 88,000 psi to 45,000 psi when a collar was clamped to it[7]. Effects of clamping stresses are illustrated graphically

Table 5—Stress-Concentration Factors due to Fillets in Heat-Treated Steel Shafts



$\frac{r}{d}$	$\frac{h}{r}$	Stress-Concentration Factor		
		Tension	Bending	Torsion
0.1	0.5	1.54	1.43	1.22
	1.0	1.65	1.60	1.25
	2.0	1.80	1.64	1.30
	4.0	1.91	1.67	1.42
0.2	0.5	1.46	1.33	1.15
	1.0	1.54	1.42	1.17
	2.0	1.66	1.44	1.20
	4.0	1.77	1.46	1.27
0.3	0.5	1.39	1.25	1.12
	1.0	1.42	1.28	1.13
	2.0	1.55	1.30	1.15
	4.0	1.66	1.32	1.19
0.4	0.5	1.34	1.19	1.12
	1.0	1.41	1.20	1.13
	2.0	1.50	1.21	1.14
	4.0	1.58	1.22	1.15

Table 6—Stress-Concentration Factors due to Threads in Heat-Treated Steel Parts

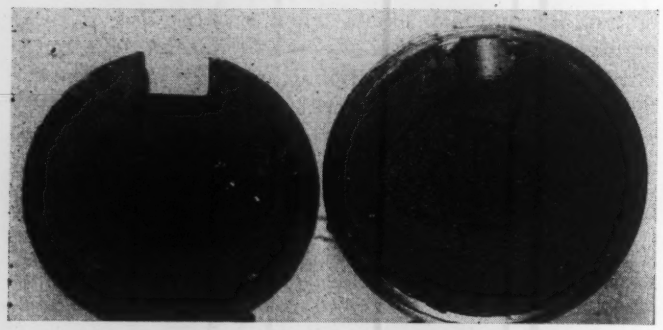
Thread System	Bending or Tension	
American National Standard	Rolled	Cut
	3.0	3.8
Whitworth		
	2.6	3.3
Dardelet		
	2.6	3.3
Aero Stud		
	2.3	2.8

in *Fig.* 67[2]. This phenomenon is further emphasized by the fact that hardened steel specimens, even though carefully and smoothly necked down to make the fracture occur at the center, often fail at the grips where the sectional area is greater than at the center. The fatigue stress-concentration factor at the grips due to clamping can quite easily be numerically greater than the ratio of the respective areas. The same problem occurs in fatigue-testing wire which usually is not necked down because of its size and because wire studies generally require that the as-drawn state be retained. There have been many railway axle failures related to press fitting the wheel on the axle. This difficulty was especially serious prior to the development of supersonic inspection methods because the fatigue cracks usually originated just within the press fit and were hard to detect.

COMBINED EFFECTS: It may often happen that a fatigue failure is the result of the combined effects of several stress concentrations. As an example, the crane shaft illustrated in *Fig.* 68 failed in fatigue at a change in section where there was a fillet with a small radius and also a keyway which terminated at the fillet. Although it would appear that the fracture originated at the junction of the keyway and the fillet, stress concentration in the fillet itself was sufficient to result in an instantaneous fracture area completely enclosed by the fatigue area. In a redesign of this shaft, the material was changed from hot-rolled mild-carbon steel to an alloy steel, a generous fillet radius was provided, and the fillet and the keyway were separated.

OVERSTRESS: The foregoing design considerations, if ignored, may often lead to fatigue failures. One more factor accounts for many failures, though it is not always attributable to the designer. This is simply overload—subjection of the machine member to loads higher than the design allowance. The condition could conceivably exist where the piece would fail in fatigue regardless of the use of high-strength steel and of all efforts to mitigate obvious stress concentrations.

The shaft fatigue failure pictured in *Fig.* 69 was one of a series of failures that occurred in a flying shear which had been partially rebuilt to cut wider and thicker stock than originally intended. Several good shafting steels were tried on replacement shafts but these, too, failed. The epidemic of failures was finally stopped by redesign of the machine to take a larger shaft.



Photo, courtesy American Cast Iron Pipe Co.

Fig. 68—Above—Fatigue failure of these identical crane shafts took place on the plane through a sharp fillet and the end of the keyway

Fig. 69—Below—This fatigue failure of a driveshaft apparently originated at the punch mark indicated by arrow

Photo, courtesy Armco Steel Corp.

Origin of Failure in Doubt

Because of this history, it was concluded that the failures had been caused principally by the overloads applied. Closer examination of the fracture face in *Fig.* 69, however, indicates that in this shaft, at least, the fatigue crack started at a surface discontinuity which appears to be a prick-punch mark. It is quite possible that the tendency toward failure could have been reduced by maintaining a smooth surface and by using a slightly less severe keyway. Although failure was assigned to overload, comparative sizes of instantaneous and fatigue zones is not such as to bear this out fully (see Parts 4 and 5 of this series, MACHINE DESIGN, Aug. and Sept.). There is also the possibility that a few unpredicted overloads might initiate the fatigue crack and that the resulting stress concentration at the root of the crack could be sufficient to promote crack growth even under normal loading.

The concluding article in this series will discuss failures resulting from defective metallurgy, fabrication, assembly, and inspection.

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Designing Tubes for Bending Loads

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FREQUENTLY the design of tubes subjected to bending is unnecessarily tedious because trial-and-error methods are used. However, a direct procedure is entirely practicable for selection of the best size of tube to carry a given bending load. It simply involves proper combination of pertinent relationships and use of a chart such as that appearing on the following page.

This data sheet explains how the lightest-weight tube can be selected by means of the chart if the maximum diameter-thickness ratio, D/t, and its relationship to modulus of rupture are known. Also outlined are simple methods for computing the margin of safety of the selected tube and the relative weights of tubes of different materials.

USE OF CHART: Trial-and-error method for tube design has been used principally because of the variation in modulus of rupture with diameter-thickness ratio. Expressed mathematically (see Nomenclature)

$$s' = K_1 s_u \quad \ldots\ldots (1)$$

where K_1 varies with D/t ratio, C. This coefficient may be evaluated from any of several satisfactory formulas or graphs*. In this data sheet it is assumed that

$$K_1 = 1.35 - 0.01C \quad \ldots\ldots (2)$$

From the common relationship

$$s = \frac{M}{Z} \quad \ldots\ldots (3)$$

it follows that

$$Z_o = \frac{M}{s'} \quad \ldots\ldots (4)$$

where Z_o is the modulus of the optimum section. The margin of safety may be defined as

$$m = \frac{s'}{s} - 1 = \frac{s'Z}{M} - 1 \quad \ldots\ldots (5)$$

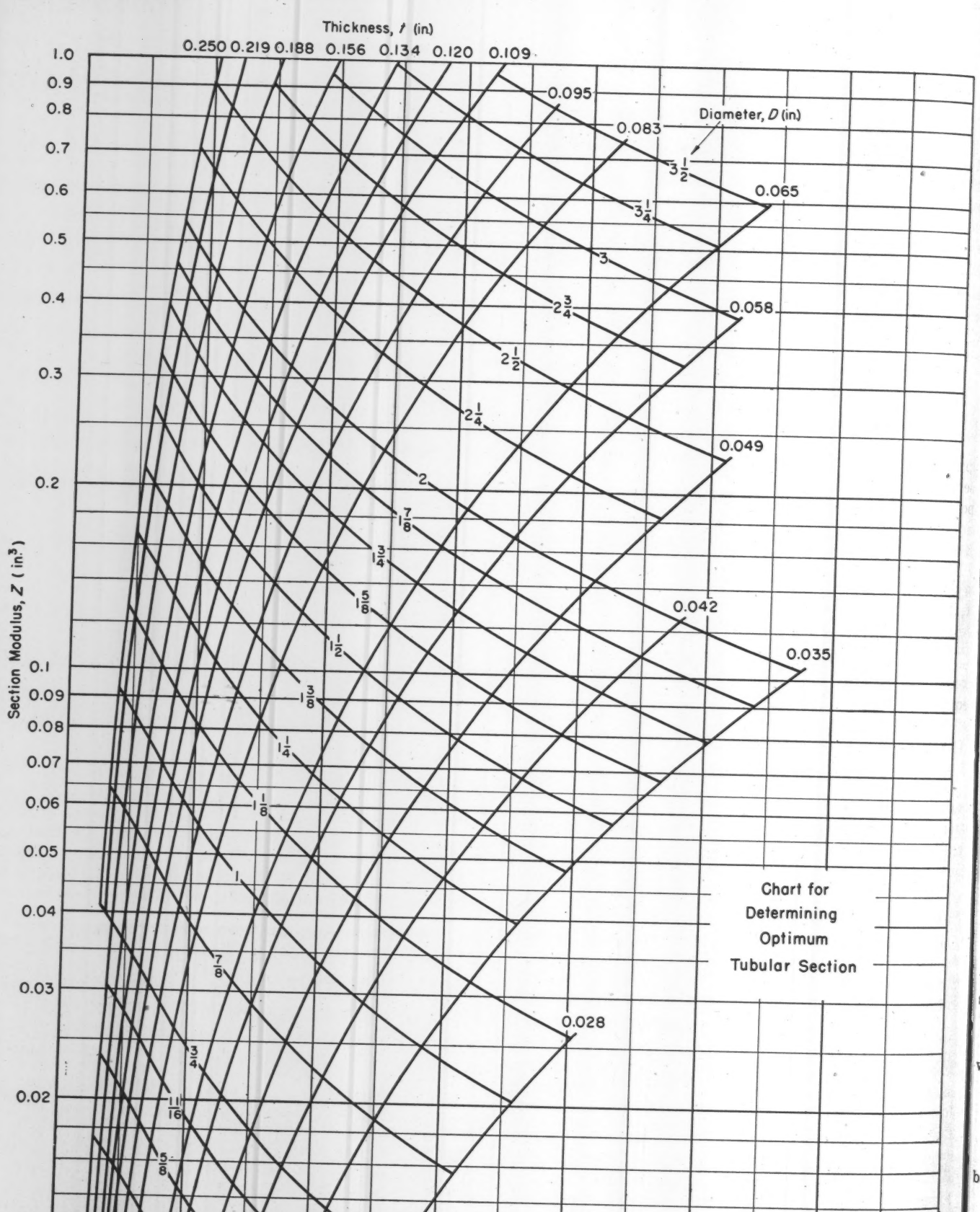
For an optimum section the margin of safety is equal to zero. If s'/f is greater than s_{yp}, yielding would take place before the limit load were reached and the *margin of safety* of most interest would be that relating to the limit load. Then, it is necessary only to use s_{yp} multiplied by the desired *factor of safety* in place of s' in Equations 4 and 5:

Nomenclature

A = Sectional area, sq in.
C = Ratio, D/t
D = Outside diameter of tube, in.
f = Factor of safety
k = Radius of gyration of tube section about transverse axis, in.
K = Constant
m = Margin of safety
M = Design bending moment, in-lb
r = Midline radius of tube, in.
s = Imposed normal stress, psi
s_u = Ultimate stress, psi
s_{yp} = Yield stress, psi
s' = Modulus of rupture in bending, psi
t = Thickness of tube wall, in.
W = Weight of tube, lb
Z = Section modulus, cu in.
ρ = Density of tube material, lb per cu in.

*Strength of Aircraft Elements. ANC-5 Handbook, U. S. Government Printing Office, Washington, D. C., 1942.

Data Sheet


Thickness, t (in.)
0.250 0.219 0.188 0.156 0.134 0.120 0.109
0.095
0.083
Diameter, D (in.)
3 1/2
0.065
3 1/4
3
0.058
2 3/4
2 1/2
2 1/4
0.049
2
1 7/8
1 3/4
1 5/8
0.042
1 1/2
0.035
1 3/8
1 1/4
1 1/8
1
7/8
3/4
11/16
5/8
9/16
1/2
0.028
Section Modulus, Z (in.³)
1.0
0.9
0.8
0.7
0.6
0.5
0.4
0.3
0.2
0.1
0.09
0.08
0.07
0.06
0.05
0.04
0.03
0.02
0.01
0
10
20
30
40
50
60
70
Diameter - Thickness Ratio, D/t = C
Chart for Determining Optimum Tubular Section

$$Z_o = \frac{M}{fs_{yp}} \quad \ldots\ldots (6)$$

$$m = \frac{fs_{yn}}{s} - 1 \quad \ldots\ldots (7)$$

The chart on the facing page shows the relationships among Z, D, t and C of a group of tubes. An optimum section may be determined by entering the graph with Z_o evaluated from Equation 4 and with the maximum allowable value of C for the material used. The best standard section is that which lies nearest, above and to the left, of the intersection of C_{max} and Z_o.

EXAMPLE: Find the optimum cold-stretched 24 ST aluminum alloy tube to carry an ultimate bending moment of 15,600 in-lb. Select the best standard section and determine its margin of safety. Assume s_u = 62,000 psi, C_{max} = 45.

From Equation 2, $K_1 = 1.35 - 0.45 = 0.90$; Equation 1, $s' = (0.9)(62{,}000) = 55{,}800$ psi; and Equation 4, $Z_o = 15{,}600/55{,}800 = 0.28$.

The intersection of $C = 45$ and $Z = 0.28$ on the graph shows that the dimensions of the optimum section are $t = 0.0575$ in. and $D = 2.59$ in. The point nearest, above and left, gives as the best standard section: $t = 0.065$ in., $D = 2.5$ in., $C = 38.5$, $Z = 0.295$.

For this standard section, $K_1 = 1.35 - 0.385 = 0.965$; $s' = (0.965)(62{,}000) = 59{,}900$ psi. Then from Equation 5, $m = [(59{,}900)(0.295)/15{,}600] - 1 = 0.13$.

CONSTRUCTION OF CHART: Coordinates of the graph may be easily calculated by means of the following equations:

$$Z = K_2 t^3 \quad \ldots\ldots (8)$$

$$Z = K_3 D^3 \quad \ldots\ldots (9)$$

where K_2 and K_3 vary with C as follows:

$$K_2 = \frac{\pi}{4} \frac{(C-1)^3 + (C-1)}{C} \quad \ldots\ldots (10)$$

$$K_3 = \frac{\pi}{4} \frac{(C-1)^3 + (C-1)}{C^4} \quad \ldots\ldots (11)$$

If it is desirable to relate the center line radius, r, to the other factors, the following similar relationships are useful:

$$Z = K_4 r^3 \quad \ldots\ldots (12)$$

where

$$K_4 = \frac{2\pi}{C} \frac{(C-1)^2 + 1}{(C-1)^2} \quad \ldots\ldots (13)$$

Equations 10, 11 and 13 are derived from a few basic relationships:

$$Z = \frac{2Ak^2}{D} \quad \ldots\ldots (14)$$

$$A = \pi t(D - t) \quad \ldots\ldots (15)$$

$$k^2 = \frac{D^2 + (D - 2t)^2}{16} \quad \ldots\ldots (16)$$

$$r = \frac{D - t}{2} \quad \ldots\ldots (17)$$

The expressions for K_2, K_3, and K_4 are developed by manipulation of these four equations.

DETERMINATION OF BEST MATERIAL: Relative weights of tubes of different materials can be easily predetermined for a given bending moment. From Equations 1 and 4,

$$Z_o s_u = \frac{M}{K_1} \quad \ldots\ldots (18)$$

The quantity M/K_1 is a constant for fixed values of M and C. For this condition, let Z_1, Z_2, s_1 and s_2 represent the optimum section moduli and ultimate tensile stresses of two different materials. Then,

$$Z_1 s_1 = Z_2 s_2 \quad \ldots\ldots (19)$$

Substituting Equation 8,

$$t_1^3 s_1 = t_2^3 s_2 \quad \ldots\ldots (20)$$

and rearranging,

$$\frac{t_1}{t_2} = \left(\frac{s_2}{s_1}\right)^{\frac{1}{3}} \quad \ldots\ldots (21)$$

Equation 21 defines the constant relationship between tubes of different materials that are optimum for given M and C.

The area of tubular section may be expressed in terms of t:

$$A = 2\pi rt \quad \ldots\ldots (22)$$

Because

$$2r = D - t = t(C - 1) \quad \ldots\ldots (23)$$

Equation 22 can be written

$$A = \pi(C - 1)t^2 \quad \ldots\ldots (24)$$

For constant C, therefore, the ratio of areas of two tubes of different materials is

$$\frac{A_1}{A_2} = \left(\frac{t_1}{t_2}\right)^2 \quad \ldots\ldots (25)$$

Also, weight is proportional to area and density, or

$$\frac{W_1}{W_2} = \frac{A_1 \rho_1}{A_2 \rho_2} \quad \ldots\ldots (26)$$

Combining Equations 21, 25 and 26,

$$\frac{W_1}{W_2} = \frac{\rho_1}{\rho_2} \left(\frac{s_2}{s_1}\right)^{\frac{2}{3}} \quad \ldots\ldots (27)$$

EXAMPLE: For constant M and C find the weight of an alloy steel tube, s_1 = 200,000 psi, in relation to that of an aluminum alloy tube, s_2 = 62,000, knowing that $\rho_1/\rho_2 = 2.83$.

From Equation 27, $W_1/W_2 = (2.83)(62{,}000/200{,}000)^{2/3} = (2.83)(0.458) = 1.296$. Hence, the alloy steel tube is 1.296 times as heavy as the aluminum alloy tube.

DESIGN ABSTRACTS

Accurate Spring Counterbalancing

By W. S. Rouverol
Lecturer in Engineering Design
University of California
Berkeley, Calif.

DESIGN of spring counterbalances for commercial purposes customarily has involved a sacrifice in accuracy for the sake of simplicity and economy. Overhead garage doors, automobile trunks, pipe hangers, and other such products are ordinarily provided with a simple form of spring loading which produces true equilibrium at some central position but 10 to 15 per cent over or under balancing at the extreme positions.

These simple spring-loading systems fail to provide equilibrium at all positions because the line of action of the load, which is usually simply the weight of the body acting downward through its center of gravity, remains parallel to itself, while its effective lever arm changes as a function of the angle of elevation. As the body is rotated, the load moment therefore must vary sinusoidally. The force exerted by a standard helical balancing spring, on the other hand, varies linearly with deflection. Reconciliation of this linear spring characteristic with the sinusoidal load moment must be accomplished if accurate balancing is to be provided.

The following analysis suggests a balancing system which appears to meet these qualifications. A standard helical spring is utilized in such a way as to produce perfect static equilibrium at all angles of elevation of the load, and the mounting of this spring is only slightly more involved than in the simplest type of direct spring loading.

In *Fig.* 1 is shown a body pivotally connected to a supporting frame at point A, about which the body may be rotated in a vertical plane; B is the point at which a load of constant magnitude and direction is applied. The load may be external, as in a pipe hanger, or may be the weight of the body itself, in which case point B is the center of gravity.

To balance the tendency of this load to cause clockwise rotation, some form of spring device, to be referred to as "the yielding means," will be attached to the body at point C, and to the stationary frame at point D. The exact location of these two points, as well as the load-deflection characteristics of the yielding means, are assumed for the present to be unknown. The direction of force F, which the yielding means exerts on the body at point C, is necessarily toward point D, however.

For perfect counterbalancing, the clockwise and counterclockwise moments about point A must be equal for all values of angle α; the equilibrium equation is then

$$W\,b\cos\alpha = F\,h \qquad (1)$$

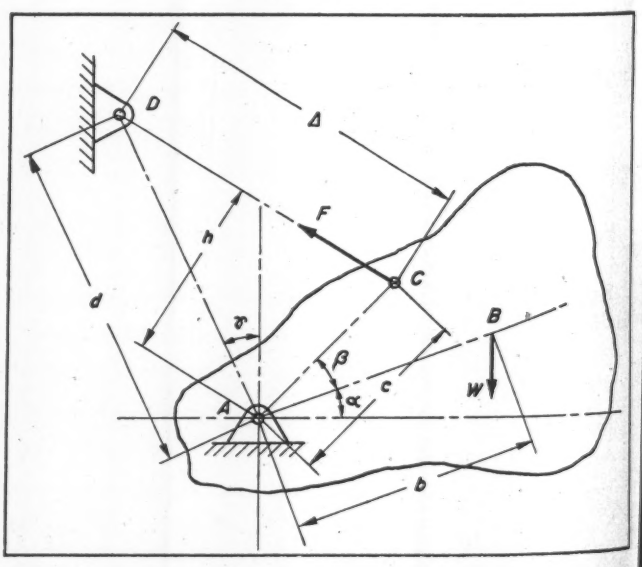
Of the three variables in this expression—angle of elevation α, spring force F, moment arm h—one can be eliminated by noting the relationship between α and h, as obtained by equating two expressions for the area of triangle ACD and simplifying

$$h = \frac{dc\cos(\alpha+\beta-\gamma)}{\Delta} \qquad (2)$$

The substitution of this value into Equation 1 gives

$$F = \frac{Wb\,\Delta\cos\alpha}{cd\cos(\alpha+\beta-\gamma)} \qquad (3)$$

Fig. 1—Free-body diagram of a counterbalanced body in static equilibrium



By proper location of point D, angle γ in this equation may be made equal to angle β. The cosine terms then cancel, leaving an expression for the spring force F in terms of a single variable, Δ

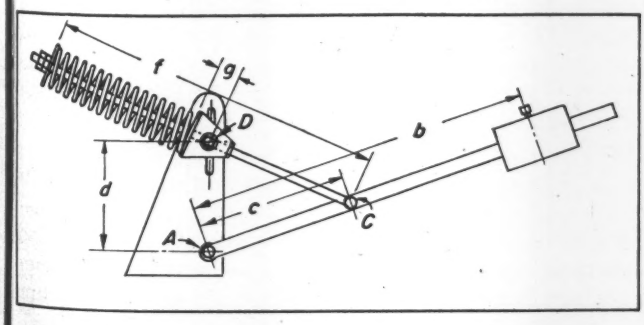
$$F = \frac{Wb}{cd}\,\Delta \qquad (4)$$

It is evident from this equation that the force F required for perfect counterbalancing is directly proportional to a linear distance Δ between the two points of attachment of the yielding means. A spring with a linear load-deflection characteristic therefore can be used, provided the force it exerts is directly proportional to the distance between its points of attachment. This requires that the distance between points C and D be at all times a direct measure of the total spring deflection. It is this requirement that prevents correct balancing merely by attachment of an extension spring between points C and D, for in this case part of the space between these points would have to be occupied by the free length of the spring.

To satisfy Equation 4, the yielding means must fulfill two conditions: it must have a spring constant equal to Wb/cd and zero initial length. The first condition is easily met, since distance d may be chosen arbitrarily. Any spring may be used which is capable of storing as strain energy the difference in potential energy of the body at its upper and lower positions. Obviously, from an economy standpoint, it will be advantageous to have a minimum of strain energy in the spring when the body is at the top position. This may be accomplished by making the arbitrary distance d as nearly equal to c as dead-center and clearance considerations permit, so that the distance Δ, and hence the force F, approach zero when the body is at top dead center.

The other condition, that the yielding means have "zero initial length," is a more unusual specification, but one which nevertheless may be met easily. For example, if a compression spring is to be used, a tie rod of effective length equal to the free length of the spring may be passed through the center of the spring, as shown in *Fig.* 2. The spring is thus constrained between a plate or washer at one end of the tie rod and a collar pivotally mounted at point D. The tie rod is passed through the collar and its free end pivotally attached at C. The collar normally will have some effective thickness g, as shown, so that the length f of the tie rod between the washers and point C must equal the free length of the spring plus the collar dimension g. Then, if the yielding means is detached for point C, and the spring allowed to expand to its free length, the end of the tie rod previously attached at C must pull back to a point immediately adjacent to point D. This is what is meant by zero initial length.

Fig. 2—*Model of a loaded member counterbalanced by a yielding means of zero initial length*



It should be noted in the construction of *Fig.* 2 that the angles β and γ of *Fig.* 1 have been taken as zero. This is of course arbitrary; the only requirement is that they be equal. *Fig.* 2 also shows a slot at D, the function of which would be to accommodate changes in the load or in the length of its lever arm, or both. For any given load and lever arm, the pivot point D would be locked in the slot at a distance d equal to Wb/ck, where k is the spring modulus. This adjustability slot is an optional feature, of value chiefly in such articles as pipe hangers to permit a single commercial model to be easily adapted to a variety of loads. It should also be noted that such an adjustability slot could be located at point C instead of D, and that the yielding means could be reversed, with the collar mounted at C instead of D, if desired. In addition, if the dimensions c and d are chosen so as to provide clearance at top dead center, the body may be completely revolved in either direction, exhibiting perfect balance at any point in the 360 degrees of travel.

From a paper presented at the ASME Semi-Annual Meeting in St. Louis, June 19-23, 1950.

Why Standards?

By G. Noble
Standards Engineer
Dominion Engineering Works Ltd.
Lachine, Que.

A *STANDARD* has been defined by Dr. John Gaillard as: "A formulation established verbally, in writing or by any graphical method, or by means of a model, sample or other physical means of representation to serve during a certain period of time for defining, designating or specifying certain features of a unit or basis of measurement, a physical object, an action, a process, a method, a practice, a capacity, a function, a performance, a measure, an arrangement, a condition, a duty, a right, a responsibility, a behaviour, an attitude, a concept or a conception." Standards fulfill several functions:

They *define* by establishing clear and proper definitions of terms so that everyone speaks the same language in this respect.

They *specify* by determining the composition and properties of materials.

They *educate* by setting targets of quality which concern both producer and consumer. They enable the consumer to buy with confidence and greatly assist the manufacturer to market a new product or design.

They *conserve* by reducing wastage, losses and the

additional tooling expense that invariably accompanies short-run or job-lot manufacturing. They promote large-scale production of standard designs and bulk purchase of materials. Savings in costs realized by sound standardization can be applied to more careful design and more precise controls which, in turn, promote further developments in efficient conservation of time and materials.

They *simplify* by reducing the number of sizes, types, grades, colors, etc., by limiting the number and variety of processes and the amount of stock and inventory. They simplify and reduce paper work and overhead expense.

They *regulate* by setting up safe and efficient operating and safety rules, by laying down proper procedures and methods for testing and inspection of products and for the proper control of quality.

They *certify* by acting as a form of hallmark of quality.

In order to appraise the value of a standard, the only true criterion is acceptance and usage. Since practically all standards are strictly voluntary in character, they must, in order to find ready acceptance by the users, fulfill certain demands. They must promote efficiency, increase production, reduce costs or improve service. If it is found that any standard, after a reasonable period of trial, has failed to obtain general adoption by the great majority of users, then it is plainly not fulfilling its proper function and should be amended or withdrawn.

Standards must be constantly reviewed, rejected or revised, and brought up to date in the light of modern progress, and to be successful, must be dynamic.

From Dominion Engineer, March, 1950.

Design by Measurement

By William T. Bean Jr.
Consulting Engineer
Detroit, Mich.

THE analytical approach to design has been oversold. It should be considered as a trial solution —not *the* solution. An engineer who thinks he can make enough assumptions to calculate the facts is lacking in experience. For example, what cyclic stress will the chosen material sustain for infinite life? For a certain finite life? In spite of our engineering heritage of thousands of tables on the strength of materials, the actual strength as found in the manufactured product will vary between thirty and three hundred per cent from the accepted handbook value. The experienced designer finds that he is confronted with some or all the following variables: size effect, surface finish, residual stress, state of stress, surface composition, homogeneity, temperature, and finally the vendor's know-how.

Mathematics is not powerful enough to disclose the true stress distribution inherent in most engineering structures. Dynamic load multipliers, due to impact and resonant vibration, are easily measured but difficult to predict. Actual service loads are found to vary from the calculated loads by a factor ranging from ten to eight hundred per cent.

In view of the prevailing errors, is the analytical approach to design outmoded by measurements? Not at all! Since the design must be built before the measurements can be taken, it will be a better initial design if based on engineering calculations instead of engineering guesses.

A fact-finding program, consisting of design development through measurement, begins when sample components are available for testing purposes. Test machines evaluate material by actually breaking the material. Nondestructive methods of quality control are established later, but must be correlated with the destructive tests. Since most machine elements are subjected to cyclical loads, their failures are predominately by fatigue. Therefore, fatigue machines are employed to obtain factual data on a material's resistance to fatigue. Constant-force fatigue machines, in which the specimen becomes a part of the resonant system, is preferred to the constant-deflection type. Test coupons, cut from the full-sized component, may be used to evaluate fatigue strength, providing there is no change in residual stress, surface finish, etc.; otherwise, the full-scale model must be broken.

Fatigue machines should be used to measure the strength of a material but not to determine the load-carrying ability of a given design. Once the allowable stress that the material will sustain is established, a stress analysis of the new design is all that is required to determine whether the allowable stresses are compatible with the operating stresses. Stresscoat is the most practical engineering tool for design analysis. The brittle-lacquer method indicates areas of high strain by fracturing at right angles to the maximum principle tensile strains. The accuracy ranges from five to twenty-five per cent, depending upon the skill of the operator. Thus, Stresscoat permits visual observation of the stress distribution and clearly defines location and orientation of wire strain gages, which have an accuracy of strain measurement of two per cent. Since wire strain gages yield "average" values over their entire length, small gages are preferred in areas of high stress concentration because more significant results are obtained.

Finding Service Loads

After stress distribution has been determined by direct strain measurement on the component itself, one unknown remains—service loads. Load analysis is best accomplished, again, by the use of wire strain gages. They may be cemented to a suitable part of the structure and calibrated in terms of load versus strain. Dynamic strains are readily converted to dynamic loads when the machine is placed in operation.

By the use of transducers employing wire strain gages, measurements of force, torque, pressure, displacement, and acceleration are readily obtained. Also available are bonded wire gages sensitive to temperature instead of strain. When these elements are

subjected to the proper phenomena, changes in gage resistance are converted into electrical energy by a Wheatstone bridge circuit. Steady as well as vibratory signals from these units may be observed on cathode-ray oscilloscopes.

We engineers are living in a wonderful age. By employing these tools of measurement, we are enabled to become as proficient as we thought we were when the boss hired us.

From a paper presented at the SAE National Tractor Meeting in Milwaukee, September 12-14, 1950.

Why Patents?

By W. R. Lane

Patent Counsel
North American Aviation Inc.
Los Angeles, Calif.

A PATENT is a contract between the government and the inventor. The inventor discloses the subject matter of the invention completely so that anyone skilled in the art can make and use it. As a reward for this disclosure, he is given the exclusive right to the invention for a limited time. After the expiration of that limited time, the invention may be used by anyone, without liability, and without compensation.

The exclusive right of the patent gives the inventor complete control over the manufacture, use and sale of the invention. He may also prevent others from making, using, or selling it during the 17-year period of the patent grant.

A patent is obtained by filing an application in the United States Patent Office, paying the fees prescribed by law, and convincing the patent examiner that the subject matter is not only patentable, but properly claimed. The patent application comprises a specification, or written description, a drawing (where the nature of the invention permits), and a claim or claims which distinguish the applicant's invention from the prior art. The claims cover only that which constitutes the applicant's contribution to the art and, by their terms, exclude that which does not belong to the applicant. The application is examined in the United States Patent Office, which maintains a corps of examiners, each skilled in his particular field. They determine the scope of the temporary monopoly to be granted to the inventor. Although the patent is granted by a government agency, and carries a presumption of validity, there is no guarantee of the rights granted. The inventor merely has a document which is supposedly enforceable by court action.

Once the patent has been acquired, the owner can feel free to invest money in the new product covered thereby without fear of competition during the life of the patent. Or he can induce others to risk capital on the basis of the temporary advantage granted by the patent. He can divest himself of his interest in the patent by assignment, or he may license others under it on an exclusive or a nonexclusive basis.

The patent owner can set the price at which patented products can be sold. Price-fixing agreements of this type are still upheld by the courts.

The sale of patented items can be restricted from a territorial standpoint. However, this control does not extend to their resale.

The licensee cannot insist that the inventor or patent owner stay out of certain fields of activity which are not covered by the patent. The courts forbid such restriction on the ground that it stifles inventive genius.

Cross licensing of patents is permitted by the courts. However, price fixing in cross-licensing agreements among patent owners or members of a patent pool is regarded as illegal. The courts presently permit cross-licensing or patent pools which are formed merely to avoid litigation or for mutual protection.

The patent owner cannot prevent use of the invention by or for the government. The act of 1910, as amended, provides that patented inventions may be used by the government and that the patent owner has a right of action against the government for such use.

The United States government is presently outlaying large sums of money for research and development. Although most of this money is for national defense purposes, there result, nevertheless, inventions having commercial uses. The usual government contract provides that the government obtain a nonexclusive, royalty-free license in any inventions emanating from the research work so as to enable the government to utilize such inventions royalty-free and without liability. The contractor can then make the benefits of the invention available to the public. In some instances the government insists upon taking title to inventions, leaving with the contractor a nonexclusive, royalty-free license. Since the government grants the patent rights, there is a serious question as to whether it can also be the holder of the patent rights. This question can be obviated if the government dedicates patents which it owns to the public, or grants royalty-free licenses to anyone upon request.

High Rate of Application Continues

The patent situation in the United States has been in a state of flux for the past twenty years. Today the Patent Office is swamped with applications. As of August 8, 1950, there were 130,392 applications awaiting action by the Patent Office. The total on file greatly exceeds that number. Patents are being granted at a rate approaching one hundred per day. Of these applications, 51 per cent are being granted to domestic corporations, 4 per cent to foreign corporations, and 45 per cent to individuals. There seems to be no abatement in the desire to disclose the invention in the hope of the reward afforded by the patent grant.

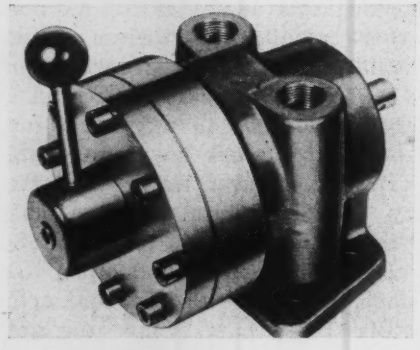
For about the last fifteen years, and until the last few months, the courts have been invalidating ap-

(*Concluded on Page* 206)

NEW PARTS

Hydraulic Pump 1

Hydropower Inc., Div. Carbide Tools Co., Milwaukee 16, Wis.



Designation: Hydropump

Style: Model P-100-A; vane type, adjustable positive displacement; manual, automatic or remote control; foot, flanged mounted or stripped

Size: 7 hp at max discharge and pressure; 1-in. pipe thread ports; 5¾-in. diameter by 9¾ in. long with 3¼-in. shaft center

Service: Hydraulic fluids; 0 to 10 gpm in either direction; 1750 rpm drive; 750 psi continuous duty, 1000 psi intermittent

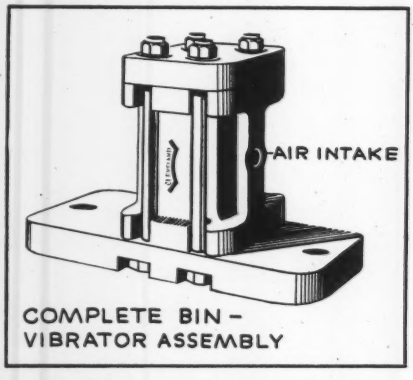
Design: Rotating pressure ring, no blade rubbing; positive vane positioning, no springs; rotary flow control for manual or automatic operation

Applications: Machine hydraulic circuits requiring control of pressure, volume and direction of hydraulic fluids without the use of valves and by-passing.

For more data circle MD 1, Page 183

Pneumatic Vibrator 2

Cleveland Vibrator Co., 2762 Clinton Ave., Cleveland 13, O.



Style: Type F, pneumatic piston

Size: Piston diameters, 1¼, 1⅝, 2 and 3 in. std.; 3 and 4 in. heavy-duty, long stroke; intake connections, ¼ to ¾ NPT; weight, 9 lb 5 oz to 185 lb

Service: Bins of lump or powdered bulk from one to several hundred tons; 750 to 1525 vibrations per minute; 8.5 to 54 cu ft air consumption at 80 psi

Design: Machined steel head and base; hard chrome-plated piston; chrome-vanadium spring; cast semisteel housing; type JN quick acting lever type valve; automatic mist lubrication; bolted mounting

Applications: Moving bulk material of lump or powder form in chutes, bins, hoppers, storage tanks, etc.

For more data circle MD 2, Page 183

Vibration Mounting 3

Robinson Aviation Inc., Teterboro Air Terminal, Teterboro, N. J.

Designation: Met-L-Flex

Style: Series 7002, all-metal cup type

Size: 2½ to 6 lb, 5½ to 12½ lb, 10 to 20 lb and 18 to 40 lb; interchangeable with #2 cupped outside load AN8008 insulator

Service: Vibration isolation and shock protection; structural strength to meet 30g drop test of Spec AN-E-19; uniformity of operation from −90C to 175C temperatures; mounts to all JAN-C-172A spec. available

Design: Nonlinear deflection characteristics for adequate load range; interwire friction for reduction of amplification at resonance and greater stability; negligible permanent set

Applications: For aircraft units subject to landing shock, gunfire recoil and combat maneuvers; vibration damping in machines, drives, etc.

For more data circle MD 3, Page 183

Bond Testing of Bi-Metallic Bearing Materials-1

THE nature of bi-metallic bearings requires close control over every step in the manufacturing process. This is not only to insure uniform quality and precision of the finished product, but, what is equally important, to insure the quality of the *bond* between the metals.

* * * *

In many bearing applications, it is desirable to combine the properties of two or more metals or alloys in order to produce a material possessing qualities not obtainable in a single metal or alloy.

The most common of these materials in use today are:

1. Lead or Tin alloys (Babbitts) on a steel back.
2. Lead or Tin alloys (Babbitts) on a bronze back.
3. Bearing bronzes on a steel back.
4. Lead or Tin alloy overlays on a bronze intermediate layer and a steel back.

In the first three cases, the soft bearing surfaces are backed up with the strength of the steel or bronze backing. In the other, the bronze layer affords some protection to the shafting in case of a failure of the overlay bearing surface, while the steel provides strength.

In all cases, their value as a satisfactory bearing material is based entirely on the strength of the inter-metallic or metallurgical bond that can be obtained between the layers. The presence and quality of this bond is assured only by constant testing and checking throughout the manufacturing process, and bond testing has become the subject of major research by the bearing manufacturers.

The ideal bond test would be quick, non-destructive and positive, however these qualifications are not met by any test yet devised. X-ray and supersonic methods have been developed which are quick and non-destructive, but unfortunately, they have not proven to be positive. Large voids can be detected by these means, but very brittle bond, weak bond, or even a complete lack of bond with intimate physical contact, will pass these tests.

Thus, the only completely positive tests available so far are of a destructive nature. Figure I illustrates two destructive tests which are used and which are quite positive—however, the amount of machining necessary for specimen preparation makes them quite slow. The most widely used test is the so-called chisel test which is quick, positive, but destructive.

Off hand this appears to be a very crude method of testing but it has been proven that, with care, accurate interpretations of the results can be made and bond quality can be kept under close control.

Figure II illustrates the chisel test in operation. The chisel is well sharpened, grinding it at an angle of approximately 10-20 degrees, and then hammered down into the material till the bond area is reached. Holding the chisel at an angle of 20-30 degrees, it is driven along the bond line. Examination of the area after chiselling will indicate the quality of the bond, strength of the bond layer and nature of the bond.

Table I lists the general types of bonding encountered with lead and tin alloys on steel and bronze backs.

TABLE I

Material	*Type of Bond*
Lead Alloy/Steel	Ductile
Tin Alloy/Steel	Brittle
Lead Alloy/Bronze	Ductile
Tin Alloy/Bronze	Brittle

The ductility of the lead alloy bond on steel and bronze is due to the fact that the alloy layer formed during the bonding process is ductile. Tin, on the other hand, forms a very brittle compound when it is alloyed with iron or copper in the bonding process. These compounds are very hard and strong in tension, but are very weak in shear strength. Thus, when a chisel is run parallel to the bond line it exerts shear force and will tend to fracture the bond through the brittle alloy layer. The lining metal curls and fractures before the chisel in a pattern which is distinctive for each type bond. From an interpretation of the tests, a semi-quantitative evaluation of the results can be obtained.

The use and interpretation of the chisel method of testing will be more fully described in the second part of this subject.

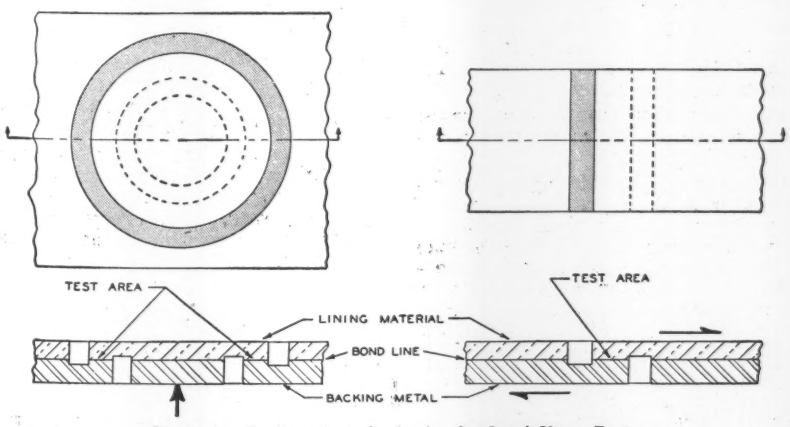


Figure I—Preparation of samples for Bond Shear Test

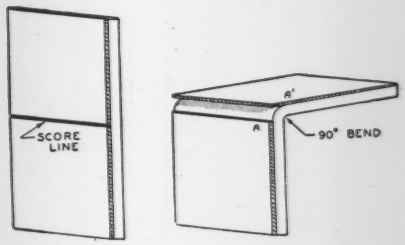
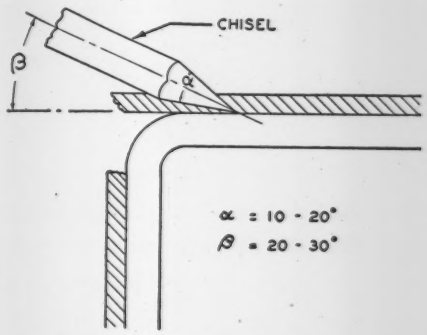


Figure II—Preparation of samples for Chisel Test



Engineering Service

Johnson Bronze offers manufacturers of all types of equipment a complete engineering and metallurgical service. We can help you determine the exact type of bearing that will give you the greatest amount of service for the longest period of time. We can show you how to design your bearings so that they can be produced in the most economical manner. As we manufacture all types of Sleeve Bearings, we base all of our recommendations on facts free from prejudice. Why not take full advantage of this free service?

This bearing data sheet is but one of a series. You can get the complete set by writing to—

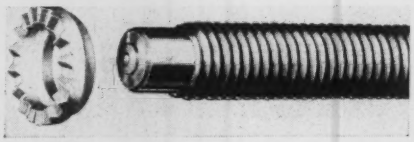
Johnson Bronze

SLEEVE BEARING HEADQUARTERS
525 S. MILL ST. • NEW CASTLE, PENNA.

NEW PARTS AND MATERIALS

Arc Welding Stud 4

KSM Products Inc., 6512 Park Ave., Merchantville 8, N. J.



Form: Roll threaded or special headed, collar, etc.

Size: ¼ in.-20 x ¾ in. to 3 in. long up to and including ½ in.-13 x 1 in. to 3½ in. long in increments of ¼ in. length and 1/16 in. diameter

Service: General fastening, Class 2 fit or better

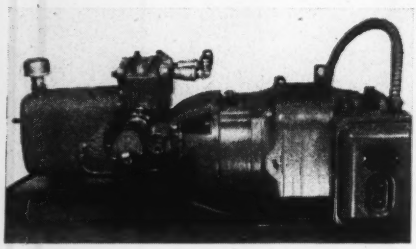
Properties: AISI C-1020 steel; ultimate strength, 65,000 psi min; yield strength, 45,000 psi min; controlled flux in each stud

Applications: For stud fastenings to be arc welded on such assemblies as chassis-frames, bumper guards, low-pressure household boilers, etc.

For more data circle MD 4, Page 183

Hydraulic Power Package 6

Harco Industries Inc., Rochester 12, N. Y.



Styles: Manual, electropneumatic or hydraulic control; three or four-way valve or combination

Sizes: 118 models, ⅓ to 5 hp, 1.8 to 8.25 gpm

Service: Hydraulic fluids to 1000 psi

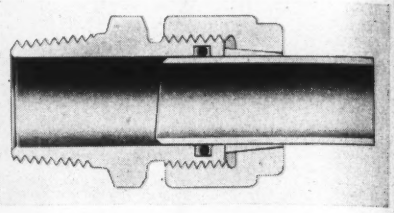
Design: Direct-connected vane type pump, totally enclosed motor, solid coupling, oil cooler, air filter, overload valve, pressure regulating valve

Applications: Hydraulic power package for a wide variety of drives and localized or single-purpose operations on machines.

For more data circle MD 6, Page 183

Tube Coupling 8

Lenz Co., Dayton 1, O.



Style: Elbows, tees, connectors and unions, male and female

Size: Tube O.D., ⅜ to 1¼ in.; pipe thread, ⅛ to 1¼ in.

Service: Hydraulic fluids to 2500 psi, 275 F

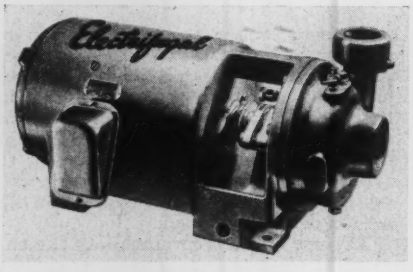
Design: O-ring seal, taper sleeve grip ring; seal and grip separate; no flares or ferrules; exact tube length not necessary for union; steel, cadmium plated

Applications: General hydraulic installations.

For more data circle MD 8, Page 183

Pump-Motor Unit 5

Allis-Chalmers Mfg. Co., 1001 S. 70 St., Milwaukee, Wis.



Designation: Electrifugal

Styles: Pump—screw connection, removable casing; flange connection, removable cover plate; motor—dripproof, splashproof, totally enclosed, explosionproof, fan-cooled

Sizes: ¾ to 10 hp

Service: Water; capacity 10 to 500 gpm, heads to 220 ft

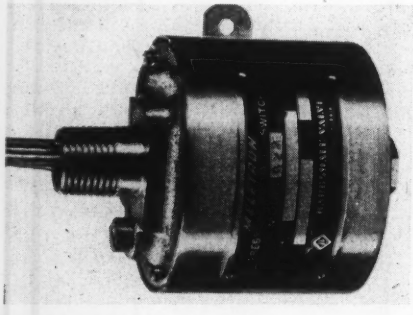
Design: Single shaft, unit cast frame; bronze wearing rings; special shaft steel; deep stuffing box; die-cast aluminum motor rotors; double-dipped, double-baked stators

Applications: General-purpose pumping.

For more data circle MD 5, Page 183

Pressure Switch 7

Barksdale Valves, 4905 Santa Fe Ave., Los Angeles 58, Calif.



Style: Model 427, pressure differential

Size: Housing—3-in. diameter, 4⅛-in. overall length; weight, 12 oz; pressure connection, ⅛-27 internal pipe thread; electrical connection, ½-in. conduit

Service: Six classes, adjustable ranges from 3 in. water to 80 psi; proof pressures to 120 psi; make or break at preset point

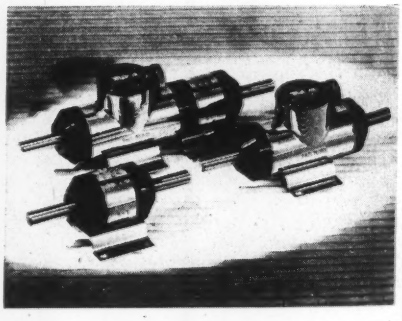
Design: SPDT ac and dc, diaphragm actuated; two pressure ports; oil-sealed housing

Applications: Maintaining liquid level with respect to liquid surface pressure; maintaining constant pressure with respect to reference other than ambient; sensing pressure difference on such as blower, ventilating and heating ducts.

For more data circle MD 7, Page 183

Miniature Speed Changer 9

Metron Instrument Co., 432 Lincoln St., Denver 9, Colo.



Style: Combination variable and fixed-ratio drive

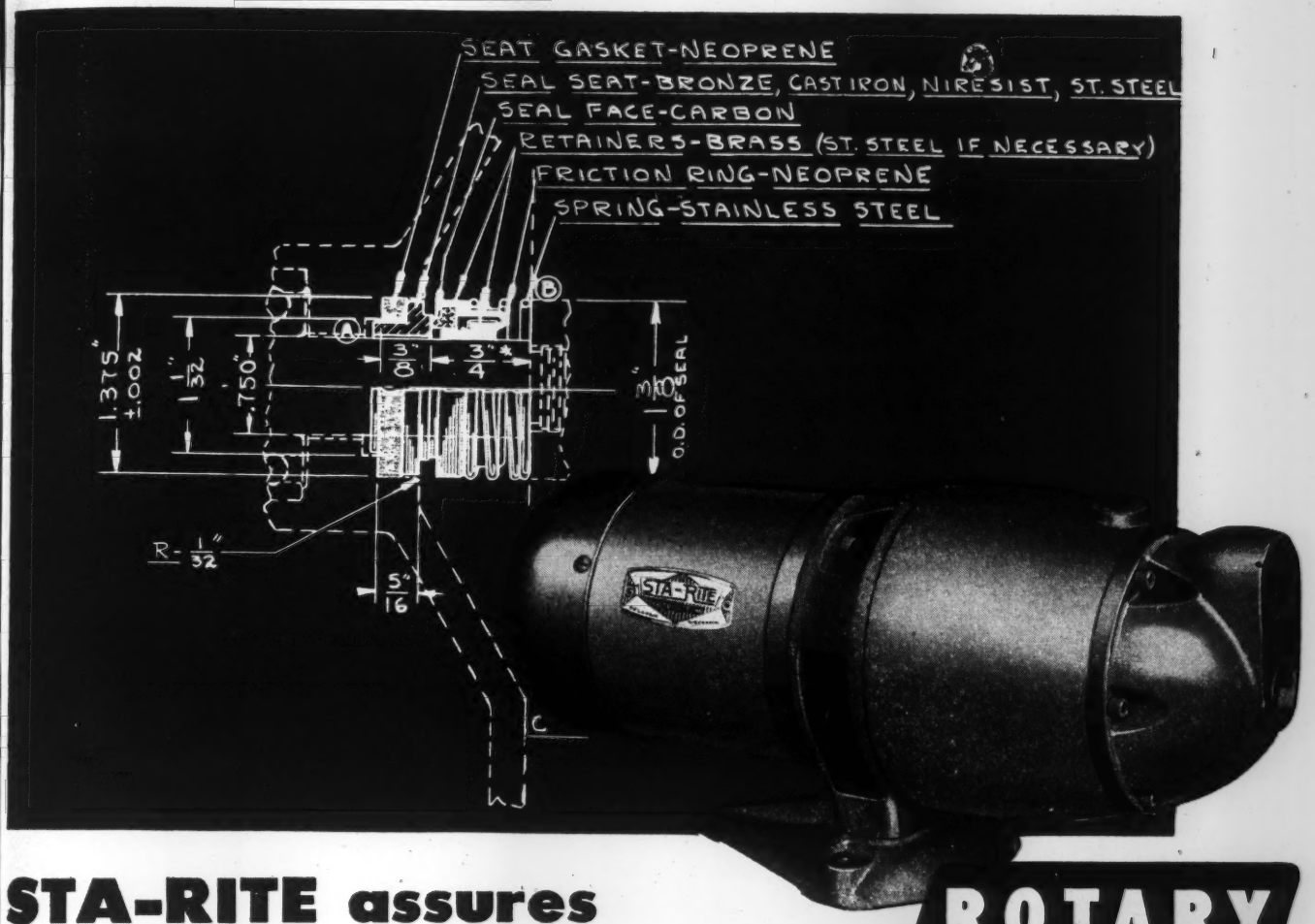
Size: Torque, 2 lb-in.

Service: Variable drive infinitely adjustable from 1/6 to 6 times fixed-ratio output; fixed-ratio drive in many standard ratios, i.e., 2:1 to 531,441:1; combination gives wide range of speeds

Design: Ball bearings; permanent lubrication; fixed-ratio units have extruded or hobbed gears

Application: Drives for timers, recorders, controllers, computers, indicating mechanisms and other low-power units requiring both speed reduction and variable speed.

For more data circle MD 9, Page 183



STA-RITE assures *Shaft-Sealing Certainty* with ROTARY SEALS

The Perma-Prime Jet Pump featured by STA-RITE Products, Inc., of Delavan, Wis., is built to give day-in, day-out operation during a long life on farms and in many other remote locations, far from service men. So every detail of this efficient, well designed unit must be just about trouble-proof. At the heart of the pump, Sta-Rite uses Rotary Seals to assure certainty in the Shaft Seal for the life of the unit.

Wherever use conditions for equipment demand *certainty* in Shaft Sealing, you'll find Rotary Seals will do the job. For STA-RITE, a special adaptation was made of the basic Rotary Seal principle which has solved tough sealing problems for manufacturers in many fields. Rotary Seal engineers can do the same for you—and, if called in at the drawing-board stage, can often suggest the simplest design approach from a Shaft Sealing standpoint, based on the wide variety of our successful experience. Write us today!

THE ROTARY SEAL PRINCIPLE

is the original approach to a practical solution of a universally troublesome problem. Our booklet "SEALING WITH CERTAINTY" explains and illustrates the principle. We're glad to send it to you without obligation.

THE ROTARY SEAL COMPANY
mechanical seals for rotating shafts

MEN OF MACHINES

Donald C. McDonald

Karl A. Panitz

Robert F. Golden

Donald C. McDonald has been appointed chief engineer of the Cook Research Laboratories, a division of the Cook Electric Company of Chicago. Before joining the Cook organization, Mr. McDonald was research supervisor of the Aeronautical Research Center, University of Michigan. Mr. McDonald is a graduate of Massachusetts Institute of Technology. His experience covers many years of professional practice in the development of servomechanisms, control systems, instrumentation, etc. In his new position, he will be responsible for all research projects.

♦

Karl A. Panitz has assumed new duties as chief engineer of National Motor Bearing Co. Inc., and its two subsidiaries, Arrowhead Rubber Company, and National Seal Co. Mr. Panitz is a graduate of the Institute of Technology in Hanover, Germany, and in 1927 came to America as assistant chief engineer for U. S. Colloid Mill Corp. later transferring to the Sperry Gyroscope Co. Inc., where he was responsible for design and layout work on gyroscopic, aviation and gunfire control instruments.

♦

Reliance Division of the Eaton Manufacturing Co. announces the appointment of **Robert F. Golden** as Works Manager. Mr. Golden joined the Reliance Division in May, 1946, and has devoted his time to development and research. Previous to joining the Reliance Division, Mr. Golden was assistant to the president of The National Lock Washer Company where he had been in succession factory manager of National's plant in Riverside, N. J., and an engineer in the Merchants Mill of the Pennsylvania Steel Co. (now Bethlehem Steel Co.)

♦

E. W. Bliss Co. announces the return of **E. V. Crane** to its engineering staff. Mr. Crane was formerly with Bliss for twenty-five years. He left during the war to accept a position with Sam Tour & Co. as vice president in charge of mechanical engineering. Later he was chief engineer for the Hydraulic Press Mfg. Co.

♦

William A. Arnken has been named president of Ardec Inc. A mechanical engineer, Mr. Arnken was associated with the Baldwin Locomotive Works from 1928-1945 and served in several capacities, including supervisory engineer in the Diesel Engineering Division. He left Baldwin in 1945 to accept the position of chief engineer for the American Precision Products Corp. From 1948 until his new appoint-

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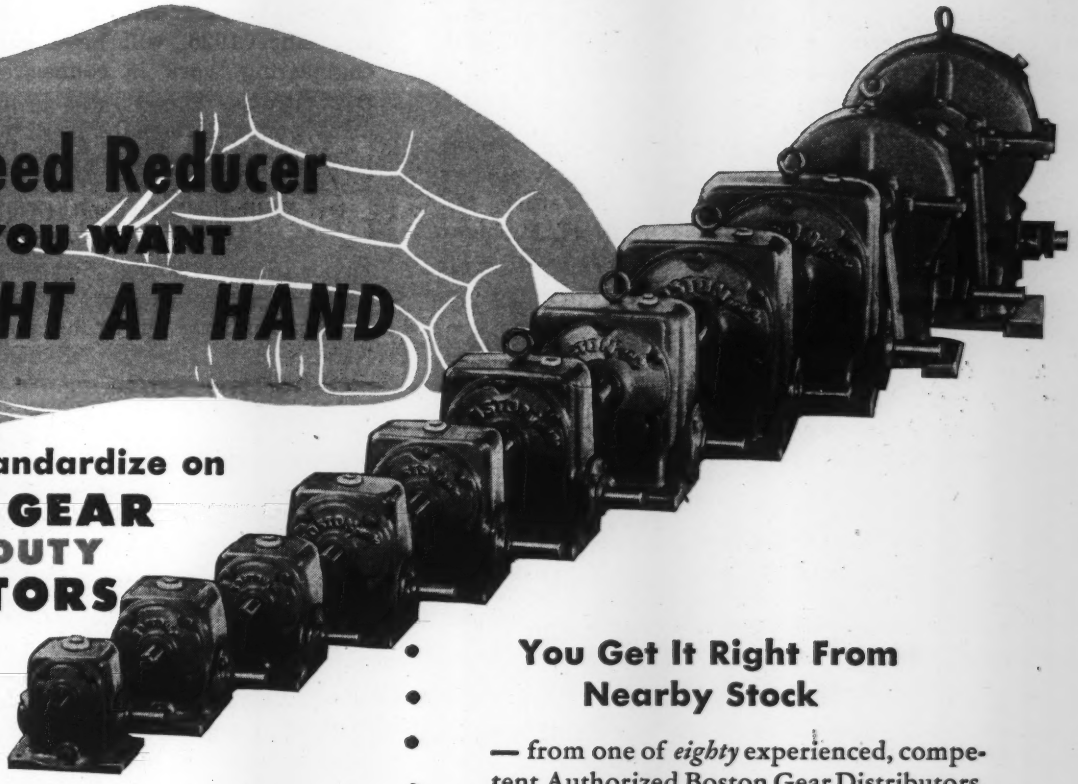
BOSTON GEAR HEAVY DUTY REDUCTORS



You Get the One Best Unit For the Job

BOSTON GEAR HEAVY DUTY REDUCTORS OFFER YOU:

1. *The widest selection of types and sizes.* Pictured above is but one (T Series, Horizontal Right Angle Drive) of *eight* types — each in a range of sizes.
2. *More trouble-free horsepower per dollar.* Heavy gearing of BOSTON *Gear* quality — worm integral with shaft-housings of Boston Gear iron — rugged steel shafts.
3. *More trouble-free hours per dollar* — the assured result of Boston Gear design, quality of materials, workmanship and quality control.

You Get It Right From Nearby Stock

— from one of *eighty* experienced, competent Authorized Boston Gear Distributors.

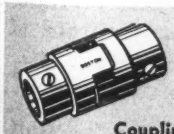
Simply consult the new Boston Gear Catalog No. 55 (copy mailed on request). It contains ready reference charts for quick selection of the one best Reductor for each and every purpose.

Ten U Series BOSTON Heavy Duty Reductors provide the right angle drive and speed reduction for the tumbling barrels in this plating tank.

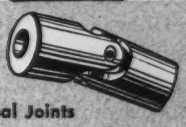
BOSTON *Gear* stocks are *Near*

BOSTON GEAR WORKS

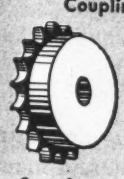
64 HAYWARD ST., QUINCY 71, MASS.



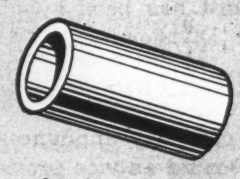
Couplings



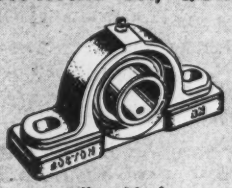
Universal Joints



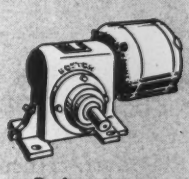
Sprockets



Bost-Bronz Bearings



Pillow Blocks



Ratiomotors

Bearings

Gears

ment, Mr. Arnken served as consultant on organization and production problems to a number of well-known industrial concerns.

♦

Acheson Colloids Corp. announces the appointment of **Paul L. Eness,** research engineer, to the staff of the company's Product Development Laboratory. Mr. Eness is a graduate of the University of Michigan and also has completed one year's graduate work in organic and protective coatings.

♦

Emil Gross has been appointed chief engineer in the hydraulic department of Allis-Chalmers Mfg. Co. **Frank Jaski,** former hydraulic engineer, becomes assistant chief engineer.

♦

E. W. Bliss Co. announces the appointment of **M. R. Hatch** as a chief engineer in its press division. Mr. Hatch's primary responsibility is development and production engineering work on large single-action mechanical presses.

♦

Alexander T. Burton has been elected vice president of North American Aviation Inc.

♦

Laurence H. Carr has been appointed director of engineering and research for Edward Valves Inc. In his new position Mr. Carr will be responsible for consolidating the development functions of engineering research, product engineering and technological specifications for process control.

♦

General Electric Co. announces the appointment of **S. A. Durbin** as product planner within the Air Conditioning Department of the Bloomfield, N. J., plant.

♦

Sherman S. Cross has been appointed operations manager of Brainard Steel Co. Mr. Cross studied engineering at the University of Akron and Goodyear Industrial University of Akron, O. He came to Brainard in 1949 as chief product engineer.

♦

George H. Garraway, director of engineering, has been elected vice president in charge of engineering and production at Orr & Sembower Inc.

♦

William W. Brown, assistant chief engineer, Sharp & Dohme Inc., has been appointed chief design engineer. Mr. Brown will supervise and direct the design and preparation of plans and specifications for equipment and facilities.

♦

E. F. Schweller and **J. L. Gibson** have been named assistant chief engineers of the Engineering Department of Frigidaire Division of General Motors. Mr. Schweller who has been with Frigidaire since 1926, will organize and head a new Research and Future Products department. Mr. Gibson, also with Frigidaire since 1926, will have general supervision over engineering work in connection with household refrigerators, commercial and air conditioning equipment, and materials and processes.

♦

Dr. William R. Osgood, former structural engineer for the David Taylor Model Basin and materials engineer of the National Bureau of Standards, has been named director of the department of mechanics at Illinois Institute of Technology. Dr. Osgood is the author of numerous technical papers in mechanics and structural engineering.

♦

Charles E. Wilson, president, General Motors Corp., will receive the American Society for Metals 1950 Medal for the Advancement of Research during the 32nd annual National Metal Congress and Exposition in Chicago.

♦

Thomas Barish, consulting engineer, has moved his office from Washington, D. C., to Shaker Heights, Ohio. Mr. Barish's present activity is in ball and roller bearings, gearing and gear applications, aircraft propellers, and machine design both practical and theoretical. Among his previous connections Mr. Barish has served as assistant chief engineer with the Marlin Rockwell Corp. and manager of the Propeller Division of the Engineering and Research Corp.

♦

Ben F. Welte, formerly assistant chief engineer of the Colonial Broach Co., has been appointed chief engineer. **Glen Harmon** succeeds Mr. Welte as assistant chief engineer.

♦

Taylor Instrument Companies announces the appointment of **William I. Caldwell** as director of research. Mr. Caldwell is a graduate of Indiana University. He was on the faculty of Purdue University for seven years, during which time he performed research on electron diffraction.

♦

L. A. March has been appointed manager of the engineering division of General Electric's Small and Medium Motors Divisions.

♦

Robert C. Juvinall, former supervisor of the engineering staff of the Chrysler Corporation research design department, has been named associate professor of mechanical engineering at Illinois Institute of Technology.

♦

John H. Howard has been appointed to the staff of the Research Division of the Burroughs Adding Machine Co.

♦

Hammel-Dahl company announces the appointment of **Maurice MacDonald** as vice president and manager of engineering.

HERE'S THE ULTIMATE IN

AIR CYLINDER POWER

• • •

The BELLOWS AIR MOTOR

Built-in Electrically-Controlled Directional Valve

•

Built-in Piston Rod Speed Controls

•

Requires only a Single Air Connection

Model illustrated — Electrically controlled BEM-5-40 2½" bore, 4" stroke air motor.

The Bellows Air Motor combines air cylinder, electrically controlled air-powered directional valve, and two independent piston speed controls all in one integral power unit . . . a unit that requires only a single air connection which can be made with flexible hose.

Here's compactness and flexibility to the N'th degree. The Bellows Air Motor makes designing for electrically-controlled air operation simple. No bulky power solenoids. No cumbersome, clumsy piping. No complex air circuits.

The unique Bellows Electroaire valve gets away from bulk and weight by using the air-power itself to shift the valve, tiny built-in 8-volt solenoid controls to direct it. Makes the whole set-up safe from electrical hazard, safe to operator, safe to machine. The Electroaire valve will operate submerged, flooded with coolant, piled high with chips or dust. And the tiny solenoid control units are guaranteed against burnout!

Integral construction means *quick, responsive power, controlled power, safe power.* For any air cylinder operation the Bellows Air Motor offers maximum effectiveness, minimum over all cost.

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Manufacturers of "Controlled-Air-Power" Devices, including conventio work feed tables, air-powered machine tool feeds, air-operat

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Mac-it Hollow Pipe Plugs — for accuracy, safety, uniformity and strength.

BETTER, FASTER SERVICE WITH THIS *COMPLETE* MAC-IT LINE!

Because many standard types of Mac-its are stocked throughout the country for quick delivery, and because specials can be engineered to your own specifications, you'll find it pays to investigate Mac-its *first.*

Mac-it's 35 years' experience in the manufacture of heat-treated, alloy steel screws is your assurance of precision, uniformity and strength. Sold through leading industrial distributors from coast to coast and in Canada. Write for new catalog today!

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THE ENGINEER'S LIBRARY

Aluminum and Magnesium Design and Fabrication

By R. Burt Schulze, supervisor of manufacturing research, Glenn L. Martin Co.; published by McGraw-Hill Book Co. Inc., New York; 589 *pages,* 6 *by* 9 *inches, clothbound; available through* MACHINE DESIGN, $7.50 *postpaid.*

This coverage enables designers, engineers and manufacturers to judge with accuracy when and where aluminum or magnesium is suited to a specific need. Particular attention is paid to machining, to welded and bonded fastenings, and to finishing. Much of the data—arranged in graphical or tabular form—is derived from the results of the recent swift-paced light-metals development program.

The opening chapters are concerned with the properties and forms of the materials. A theory of forming is established involving basic stress and strain, progression of strain in work hardening, and types of stresses related to forming problems. The following production and fabrication methods are discussed: machining; forming in experimental production, limited production, and mass-production; mechanical and welded fastening. The machines and tools of each method are described comprehensively and discussions of their limitations and how these effect product design are included. An important phase of production and fabrication—cost analysis—is also presented with sufficient tabular data to serve as a guide to actual studies. Summarized in the concluding chapters are data relevant to the design principles and applications of aluminum and magnesium metals.

Manufacturing and Association Publications

Strength of Stainless Steel Structural Members as Function of Design: An excellent compilation of data on the basic properties of light gauge stainless strip of various tempers at room temperature and a design procedure for calculating the allowable loads of structures made from such strip. Presented in a well-organized manner adequately supported by graphical data are discussions of: stress-strain curves of cold-rolled high tensile stainless steel strip as compared to other metals; influence of the elastic characteristics of cold-rolled strip on the determination of allowable stresses; basic tension and compression data; combined moduli for use in compression; bending; shear buckling of flat and curved sheets; torsional instability of open section columns. Throughout, examples are used to illustrate the application of data. This 153-page, 7½ by 10½-in., clothbound book was

Honing machine with a 96-foot stroke gets precision with TIMKEN® bearings

WHEN honing the inside of a long precision tube, a smooth-running spindle with steady, precise rotation is of major importance. That's why the Barnes Drill Company uses Timken® bearings on this No. 36 horizontal honing machine built for the Watervliet Arsenal. Eight Timken bearings are used in the headstock.

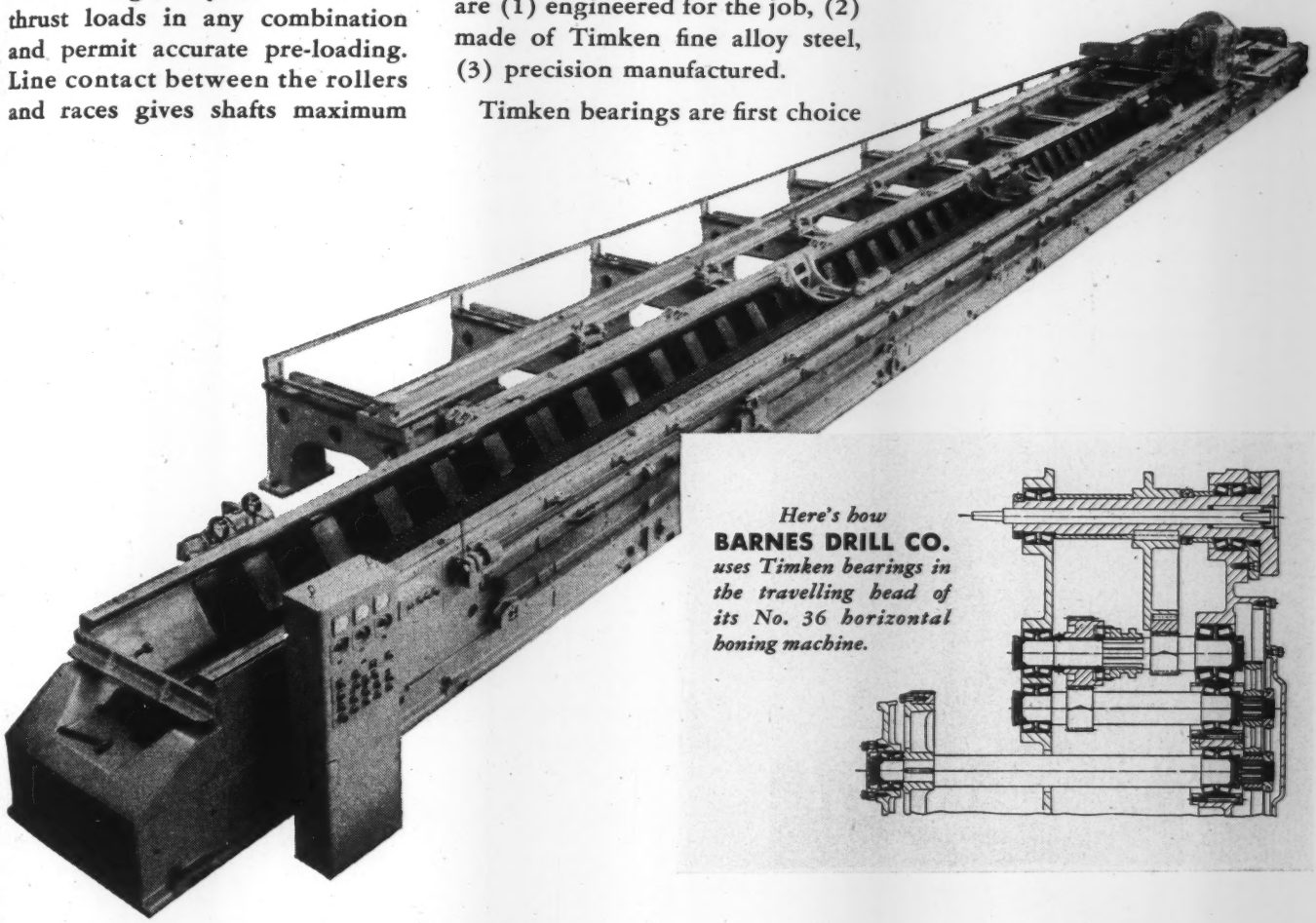
Due to the tapered design of Timken bearings, they take radial and thrust loads in any combination and permit accurate pre-loading. Line contact between the rollers and races gives shafts maximum support. Gears mesh correctly, vibration is eliminated and wear on moving parts is prevented. Incredibly smooth surface finish and true rolling motion practically eliminate friction. And because Timken bearings permit tighter closures, maintenance and lubrication are cut to a minimum.

Timken bearings normally last the life of the machine because they are (1) engineered for the job, (2) made of Timken fine alloy steel, (3) precision manufactured.

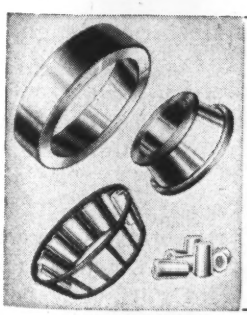
Timken bearings are first choice throughout all industry. No other bearing can give you *all* the advantages you get with Timken bearings. Look for the trade-mark "Timken" on every bearing you use. The Timken Roller Bearing Company, Canton 6, Ohio. Canadian plant: St. Thomas, Ontario. Cable address: "TIMROSCO".

TIMKEN BEARING EQUIPPED

This symbol on a product means its bearings are the best.



Here's how
BARNES DRILL CO.
uses Timken bearings in the travelling head of its No. 36 horizontal honing machine.



DESIGN LEADERSHIP

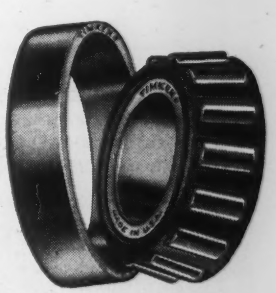
The first Timken tapered roller bearing was produced in 1898. Since then the one-piece multiple perforated cage, wide area contact between roll ends and ribs, and every other important tapered roller bearing improvement have been introduced by The Timken Roller Bearing Company.

The Timken Company leads in: 1. advanced design; 2. precision manufacture; 3. rigid quality control; 4. special analysis steels.

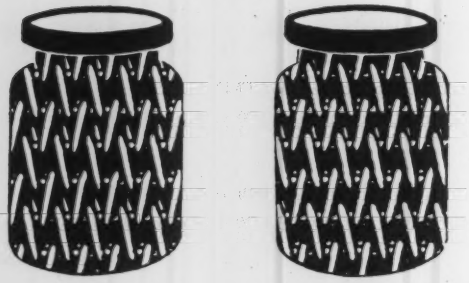
TIMKEN

TRADE-MARK REG. U. S. PAT. OFF.

TAPERED ROLLER BEARINGS



NOT JUST A BALL ◯ NOT JUST A ROLLER ▭ THE TIMKEN TAPERED ROLLER ▭ BEARING TAKES RADIAL ◫ AND THRUST →◫← LOADS OR ANY COMBINATION →✳←



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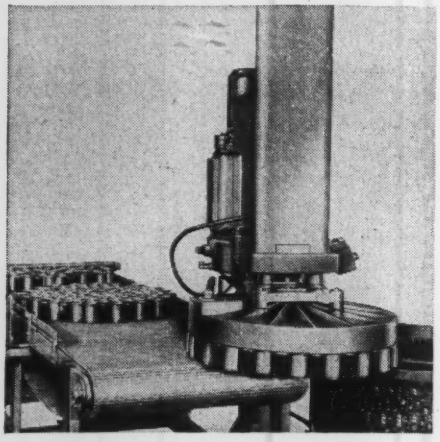

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prepared by Michael Watter, director of research, The Budd Co. and R. A. Lincoln, Allegheny Ludlum Steel Corp. Copies may be obtained from the Allegheny Ludlum Steel Corp., Pittsburgh, Pa., for $5.00 each.

Principles of Stretch-Wrap Forming: This 87-page, 8½ by 11-in., clothbound book presents the theory, advantages and uses of stretch-wrap forming equipment in modern aircraft manufacture. Well-illustrated and terse, the discussions include detailed tooling procedures and form die construction data. Copies may be obtained from the Hufford Machine Works Inc., 207 N. Broadway, Redondo Beach, Calif., for $5.00 each.

How To Use Kodagraph Reproduction Materials: A detailed description of Kodagraph silver-sensitized reproduction papers. The scope includes properties, applications and techniques. Copies of the 72-page, 5¾ by 8½-in., paper-bound booklet may be obtained from the Eastman Kodak Co., Rochester 4, N. Y., for $0.50 each.

Government Publications

Modes and Frequencies of a Modified Structure—NACA TN 2132: A method is developed for the calculation of the natural coupled or uncoupled frequencies and modes of a structure with modifications, such as the addition of concentrated masses or springs, directly from the modes and frequencies of the unmodified structure. The modes of the modified structure are expanded in terms of the modes of the unmodified structure. A characteristic equation and a frequency determinant are derived by the use of the Galerkin method. Numerical examples are presented to show the accuracy of the method and the number of modes and frequencies of the unmodified structure necessary for agreement with exact solutions. Copies of this 23-page report may be obtained by writing to the National Advisory Committee for Aeronautics, 1724 F Street, N.W., Washington 25, D. C.

Turning-Angle Design Rules for Constant-Thickness Circular-Arc Inlet Guide Vanes in Axial Annular Flow—NACA TN 2179: Data obtained from tests of several conventional axial-flow compressor guide-vane designes in annula cascade are presented. A linear relationship between air turning angle and vane-camber angle for the vane setting of zero angle of incidence was found for vanes in convergent annuli for the conditions of solidity of 1.4 to 1.7, inlet Mach number of approximately 0.3, and annulus area ratio across the vanes of 0.86 to 0.95. The measured turning data were also corrected to constant inlet axial velocity in order to obtain a corrected design rule applicable over a wide range of axial-velocity ratio across the vane. Copies may be obtained from the National Advisory Committee for Aeronautics, 1724 F. Street, N. W., Washington 25, D. C.

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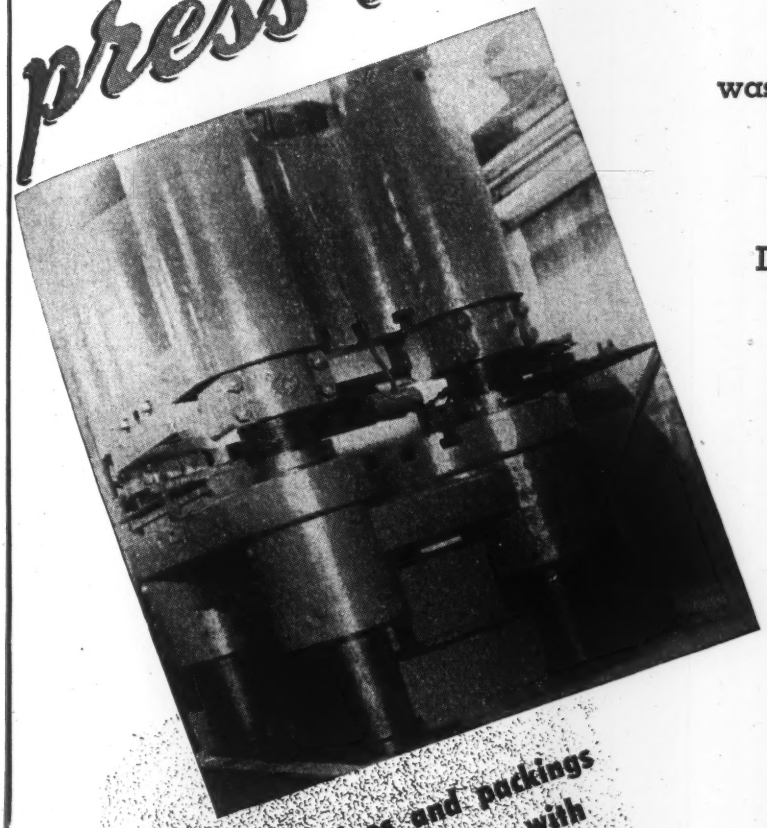

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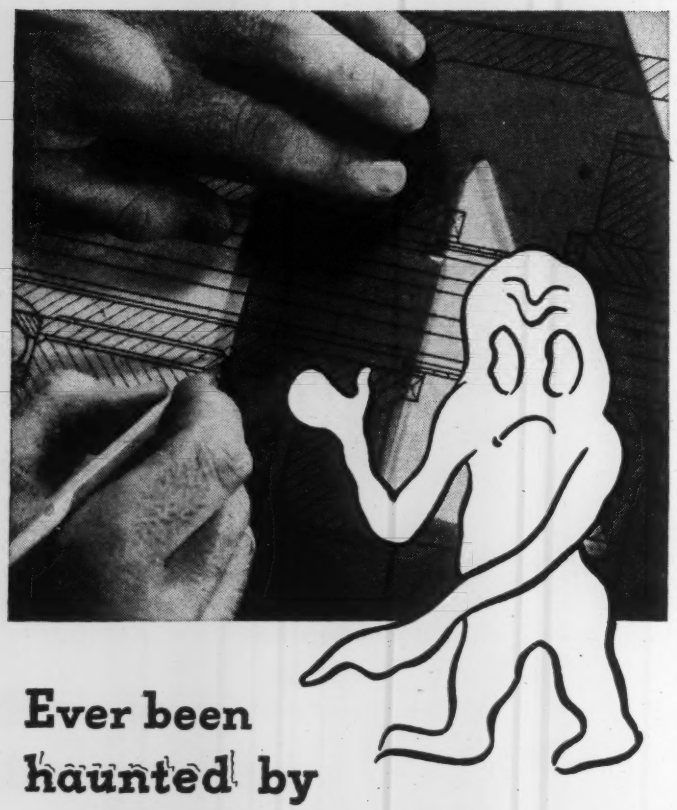

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PROFESSIONAL VIEWPOINTS

". . . determining effective pendulum length"

To the Editor:

Our attention has been called to the interesting data sheet, "Determining Radius of Gyration," by John T. Hunt in the August, 1950, issue of MACHINE DESIGN. The nomograph which forms a part of this article should be very helpful in evaluating the basic equation.

However, difficulty may be encountered in some instances in applying the equation because of inconvenience in determining the effective length, r, of the pendulum. A change in the length of the pendulum can often be determined more conveniently. If this is the case, the experiment is repeated with a different pendulum length, and the data from the two experiments are used simultaneously to determine the two lengths.

This procedure has been found useful in experimental work on aircraft where the method of laying the equipment on its side and balancing it on a knife edge obviously is impractical. This further refinement in the method of determining the radius of gyration is described by William Gracey in NACA Technical Note No. 1629, "The Experimental Determination of the Moments of Inertia of Airplanes by a Simplified Compound-Pendulum Method."

As adapted from this paper, the simplified development of the two-test procedure is as follows:

In terms of long and short suspension lengths, r_l and r_s, and corresponding periods of oscillation, T_l and T_s, the radius of gyration, k, of an object about its center of gravity can be stated by the two equations,

$$k = \sqrt{\frac{T_l^2 r_l g}{4\pi^2} - r_l^2}$$

$$k = \sqrt{\frac{T_s^2 r_s g}{4\pi^2} - r_s^2}$$

The short suspension length can be found by letting $\Delta r = r_l - r_s$, equating the two equivalent expressions for k, and simplifying:

$$r_s = \frac{4\pi^2 (\Delta r)^2 - T_l^2 g (\Delta r)}{(T_l^2 - T_s^2)\, g - 8\pi^2 (\Delta r)}$$

The long suspension length, r_l, is determined simply by adding the known value of Δr to the computed value of r_s.

With both lengths known, it is possible then to compute two values for the radius of gyration and obtain their average.

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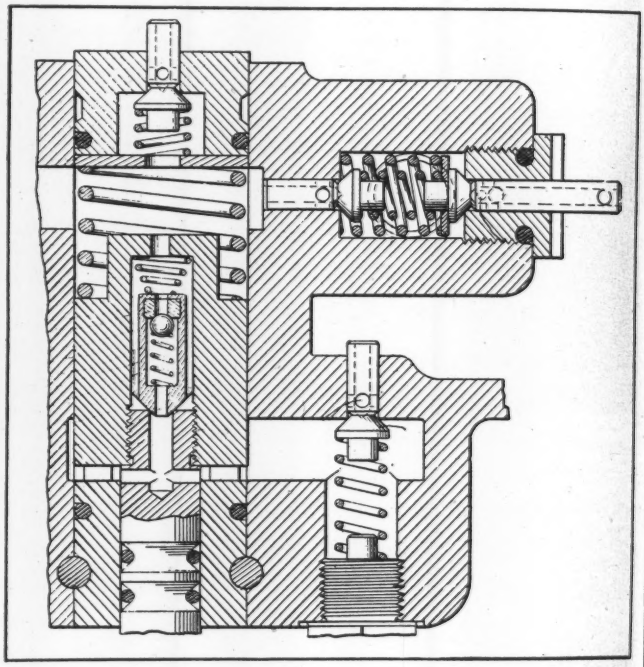
NOTEWORTHY PATENTS

VARIABLE-CAPACITY reciprocating plunger pump for small hydraulically operated equipment produces low-pressure high-volume output at low loads and automatically shifts to higher-pressure lower-volume flow when higher pressure is required. The device covered in patent 2,510,150 includes a conventional piston arrangement for low-pressure



output, with discharge volume equal to the volume displaced by the piston. When a predetermined discharge pressure is reached, a spring-loaded ball within the main piston lifts, allowing some of the oil to flow back below the piston. Volume of oil pumped under this condition is reduced to the volume displaced by the piston rod, with a proportional increase in the discharge pressure capacity. Hydraulic Equipment Co. has been assigned the patent by William T. Stephens.

CENTRIFUGAL SEPARATION of oil particles from refrigerant gas is utilized in a hermetically sealed motor-compressor unit. A small clearance between the rotor (which rotates around the stator) and the housing permits the compressed gas to be thrown radially outward, the heavier oil particles being deposited on the housing wall and draining back to the compressor. A spiral strip secured to

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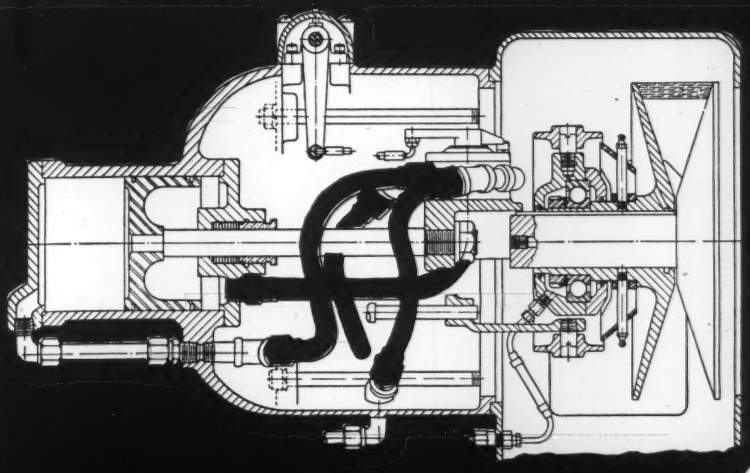

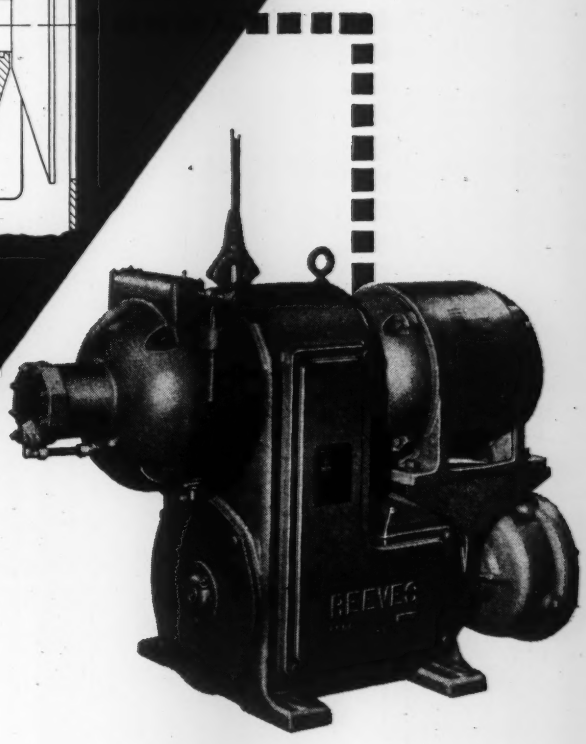

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For application Data Sheet No. 102, write C. A. Norgren Co., 242 Santa Fe Drive, Denver 9, Colorado

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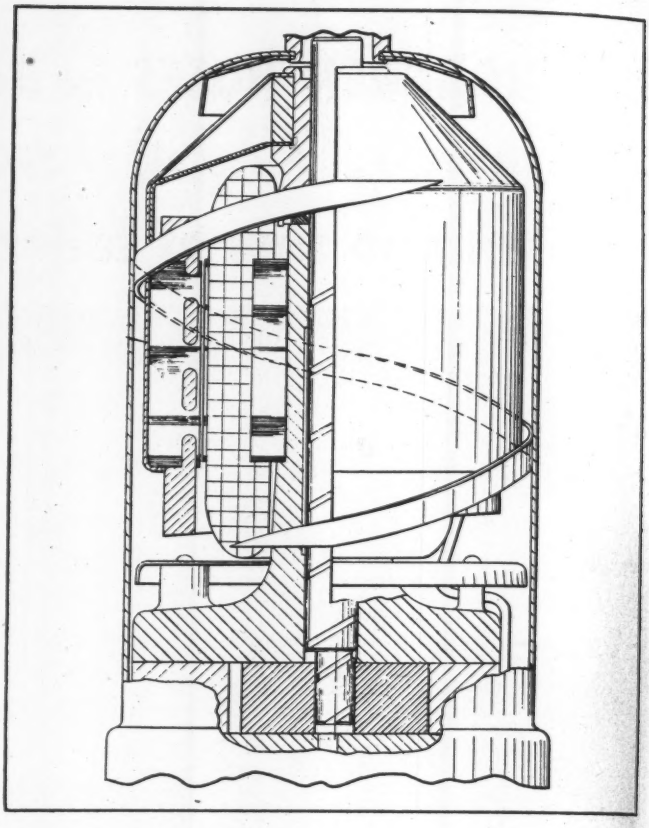
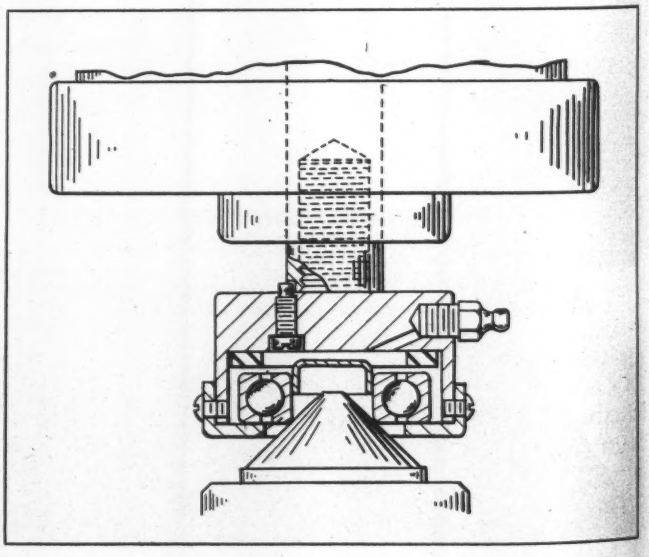


the inside of the outer housing assists in guiding the oil down the housing walls. The compressor is covered in patent 2,513,912, assigned to Jack & Heintz Precision Industries Inc. by Samuel Bousky.

FLOATING BEARING in an assembly for transmitting axial thrust between rotating and nonrotating members permits the lateral freedom usually necessary in very high speed spindle rotation, as in a centrifuge. A ball bearing, free to move both radially and axially in a cage on the stationary thrust member, engages a tapered spindle on the rotating member. When it is desired to move the



How to keep a rubber gasket in its place

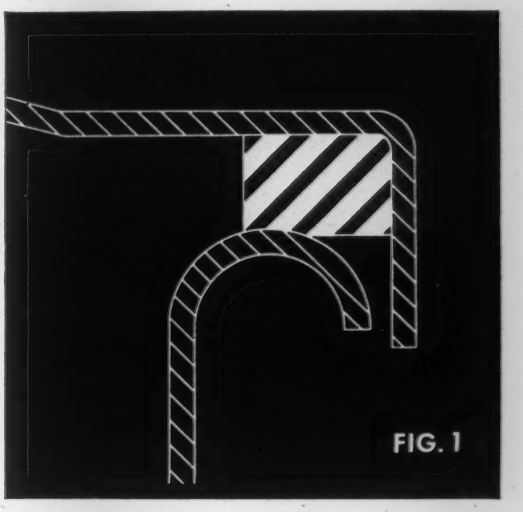


FIG. 1

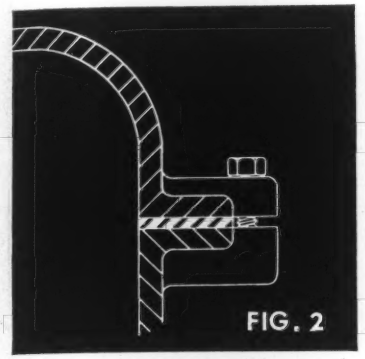


FIG. 2

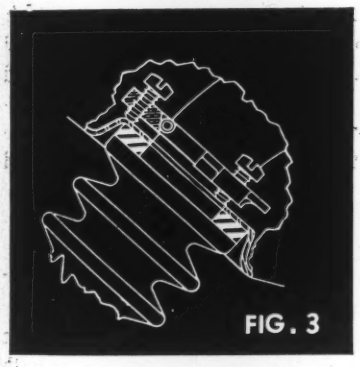


FIG. 3

For some applications, rubber makes an ideal gasket. In many cases, however, rubber's lack of compressibility introduces certain problems. Under pressure, rubber tends to creep out of place unless confined on four sides. This is especially true when lubricants are present.

The obvious way to overcome this tendency to side flow under pressure is to add to rubber the amount of true compressibility required for each application. This is done by compounding the various rubbers with calculated amounts of compressible cork particles. When such a material is compressed within its recommended range, it compresses itself, actually decreasing in volume, with virtually no tendency to creep.

Cork introduces a second factor that inhibits lateral flow, i.e., high surface friction. The surface of a typical Armstrong's Cork-and-Rubber Composition is dotted with cork particles. Since this cork retains most of its friction when wet with oil, cork-and-rubber gaskets are almost a necessity on oily flanges.

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Some typical cork-and-rubber applications are illustrated on this page. Figure 1 shows how one of Armstrong's Cork-and-Rubbers conforms to a rolled tank edge with no tendency to slip off. Figure 2 shows the flange of an oil-filled switch. Cork-and-rubber seals it under high flange pressure without creep. In Figure 3, cork-and-rubber cushions a transformer bushing and keeps it oiltight.

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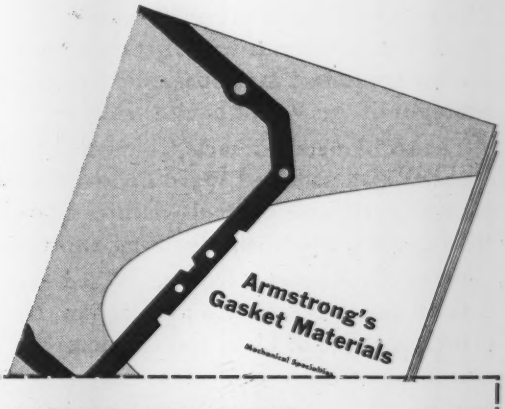

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spindle axially for control purposes, thrust is applied to the bearing outer race by a nonrotating plunger, and is transmitted to the spindle by the inner bearing race. Axial clearance between bearing and bearing cage allows the bearing to become aligned with the rotating spindle before the thrust load is applied.

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What Size Motor?

(*Continued from Page* 130)

screw gearing of very low efficiency.

From the viewpoint of motor size, certain conclusions may be drawn from TABLE 5. First, the hydraulic motor has little advantage over the electric motor in the rudder application other than its normal weight reduction which is not too important on board a ship. Second, the hydraulic motor has a pronounced advantage over the electric motor in the turret application with respect to power rating and weight.

Little additional motor power above the load requirement is necessary with a hydraulic turret drive, whereas the electric turret drive requires a motor four times as large as the load rating. The power rating advantage of the hydraulic motor in the turret application, combined with its normal advantage with respect to weight, gives it a total weight advantage of 50 to 1 over the electric motor. Because weight is so important in aircraft equipment, this means an electric turret drive of the Ward-Leonard type is out of the question unless enormous improvements are made in electrical designs.

What has been observed for the turret and rudder examples is typical of other applications. As peak load velocity to acceleration ratios dip below the order of one-half second, the hydraulic motor begins to exhibit extraordinary advantages over the electric motor with respect to size requirements.

Appendix

According to the simplifying assumptions made, two fundamental inequalities govern the choice of the gear ratio to go with a given motor and a given load. First, the peak motor torque must be greater than or equal to the sum of the peak torque required to drive the load as reflected through the gear ratio

R plus the peak torque required to accelerate the motor. Thus

$$T_m \geqq \frac{J_l a_l + T_l}{R} + J_m a_l R \quad \ldots\ldots (6)$$

Second, the peak motor velocity must be greater than or equal to the peak load velocity multiplied by the gear ratio, or

$$v_m \geqq v_l R \quad \ldots\ldots (7)$$

Multiplying Relation 6 by $v_l^2 R / J_m a_l v_m^2$, then

$$\frac{\left(\frac{v_l}{a_l}\right)}{\left(\frac{v_m J_m}{T_m}\right)}\left\{\frac{R}{\left(\frac{v_m}{v_l}\right)}\right\} \geqq \frac{\left(\frac{v_l}{a_l}\right)}{\left(\frac{v_m J_m}{T_m}\right)} \quad \frac{[v_l(J_l a_l + T_l)]}{(v_m T_m)} +$$

$$\left\{\frac{R}{\left(\frac{v_m}{v_l}\right)}\right\}^2 \quad \ldots\ldots (8)$$

Use of definitions for Equations 1 through 4 allows Relation 8 to be written as

$$\frac{1}{\left(\frac{\tau_m}{\tau_l}\right)}\left\{\frac{R}{\left(\frac{v_m}{v_l}\right)}\right\} \geqq \frac{1}{\left(\frac{\tau_m}{\tau_l}\right)} \frac{1}{\left(\frac{P_m}{P_l}\right)} +$$

$$\left\{\frac{R}{\left(\frac{v_m}{v_l}\right)}\right\}^2 \quad \ldots\ldots (9)$$

Relation 9 sets an upper and lower limit on the quantity $R/(v_m/v_l)$ and may be rewritten as

$$\frac{1}{2\left(\frac{\tau_m}{\tau_l}\right)}\left\{1 - \sqrt{1 - \frac{4\left(\frac{\tau_m}{\tau_l}\right)}{\left(\frac{P_m}{P_l}\right)}}\right\} \leqq \frac{R}{\left(\frac{v_m}{v_l}\right)}$$

$$\leqq \frac{1}{2\left(\frac{\tau_m}{\tau_l}\right)}\left\{1 + \sqrt{1 - \frac{4\left(\frac{\tau_m}{\tau_l}\right)}{\left(\frac{P_m}{P_l}\right)}}\right\} \quad \ldots\ldots (10)$$

Relation 7 may be written as

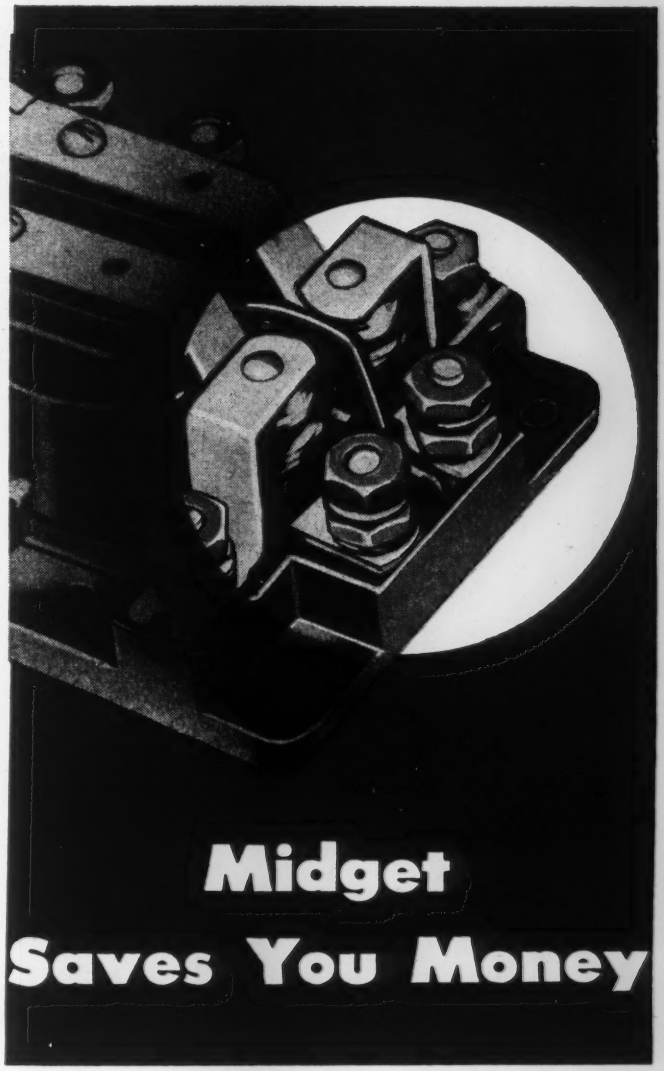
$$\frac{R}{\left(\frac{v_m}{v_l}\right)} \leqq 1 \quad \ldots\ldots (11)$$

and sets an upper limit only. The lower of the two upper limits for Relations 10 and 11 controls the value of $R/(v_m/v_l)$.

The radicals in Relation 10 must be real for a physical gear ratio. Therefore,

$$\frac{P_m}{P_l} \geqq 4\,\frac{\tau_m}{\tau_l} \quad \ldots\ldots (12)$$

This condition guarantees gear ratio clearance in Relation 10 when



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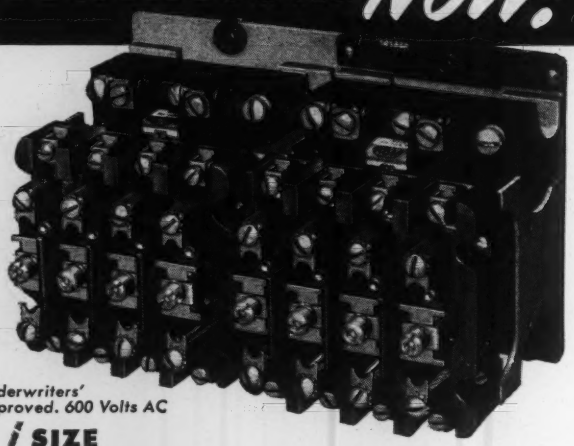

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5 to 8 Pole 5-9/16″ w. x 3-5/8″ h. x 3-5/16″ d.

Reversing

2 to 4 Pole 5-9/16″ w. x 3-5/8″ h. x 3-5/16″ d.

Note: 10 and 15 ampere contactors have same mounting and overall dimensions.

ACCESSIBILITY

To replace contacts, it is not necessary to disassemble the complete contactor. Just remove the parts comprising the stationary and movable contacts. Contacts can be replaced without disturbing wiring. To change coil, remove magnet frame and coil assembly only. (See illustration below.)

FLEXIBILITY

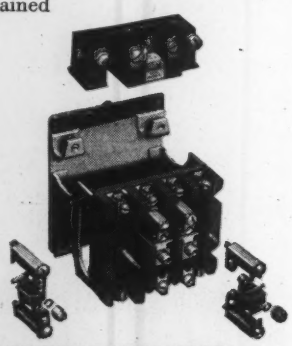
Using a screw driver only, you can easily change any pole from normally open to normally closed. No special parts required. 10 and 15 ampere parts are interchangeable.

RELIABILITY

Laboratory tests involving millions of operations, plus field service of thousands of R-B-M contactors on door operators, radio transmitters, packaging and weighing machinery, hoists, machine tools and many other industrial applications offer proof of dependable, trouble-free performance.

ADVANCED DESIGN

Melamine Insulation. Molded coil housing. Ilsco solderless connectors. 50/60 cycle magnet coils. Palladium silver contacts. Stainless steel self-contained contact springs.



Where space is a factor, and accessibility a must—use R-B-M industrial contactors. Initial low cost plus dependable performance will save you money. Write for Bulletin 600 and price list on your company letterhead.

Dept. B-11, R-B-M DIVISION OF ESSEX WIRE CORP.

RBM

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MANUAL AND MAGNETIC ELECTRIC CONTROLS
—FOR AUTOMOTIVE, INDUSTRIAL, COMMUNICATION AND ELECTRONIC USE

$$\frac{\tau_m}{\tau_l} \geqq 0.5 \quad \ldots\ldots (12a)$$

since the left-hand limit of Relation 11 is then certain to be less than unity and also less than the right-hand limit.

When $\tau_m/\tau_l \leqq 0.5$, gear ratio clearance requires the lower limit of Relation 11 to be less than unity, or

$$\sqrt{1 - \frac{4\left(\frac{\tau_m}{\tau_l}\right)}{\left(\frac{P_m}{P_l}\right)}} \geqq 1 - 2\left(\frac{\tau_m}{\tau_l}\right) \quad \ldots\ldots (13)$$

in which both sides are positive numbers. Squaring both sides yields

$$\frac{-4\left(\frac{\tau_m}{\tau_l}\right)}{\left(\frac{P_m}{P_l}\right)} \geqq -4\frac{\tau_m}{\tau_l} + 4\left(\frac{\tau_m}{\tau_l}\right)^2 \quad \ldots\ldots (14)$$

$$\frac{P_m}{P_l} \geqq \frac{1}{1 - \frac{\tau_m}{\tau_l}} \quad \ldots\ldots (15)$$

Satisfying Relation 15 when $\tau_m/\tau_l \leqq 0.5$ automatically fulfills Relation 12.

In summary, Relations 10 and 11 imply that

$$\frac{P_m}{P_l} \geqq \frac{1}{1 - \frac{\tau_m}{\tau_l}}; \text{ when } \frac{\tau_m}{\tau_l} \leqq 0.5 \quad \ldots\ldots (16)$$

or

$$\frac{P_m}{P_l} \geqq 4\frac{\tau_m}{\tau_l}; \text{ when } \frac{\tau_m}{\tau_l} \geqq 0.5 \quad \ldots\ldots (17)$$

which is the result to be proved.

The author wishes to acknowledge the valuable constructive criticism received from his colleagues in the Servomechanisms Laboratory during the writing of this article which is based partially on work sponsored by the U. S. Air Force. This article has been developed from a paper[10] by the author entitled "Comparison of Hydraulic and Electric Servomotors" presented at the National Conference on Industrial Hydraulics.

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"Catsup ON YOUR ICE CREAM?"

Just because you used catsup on your steak, the waiter shouldn't assume you use it on the rest of your dinner too!

And just because one bearing is best lubricated by one particular grade of oil, you shouldn't assume that the same oil is best for *all* bearings on that machine. In many cases it isn't.

OIL CUPS permit you to lubricate each bearing with the oil best suited to that bearing—thus prolonging bearing life, reducing maintenance costs, cutting down-time, boosting production. And oil cups fortunately *cost very little.*

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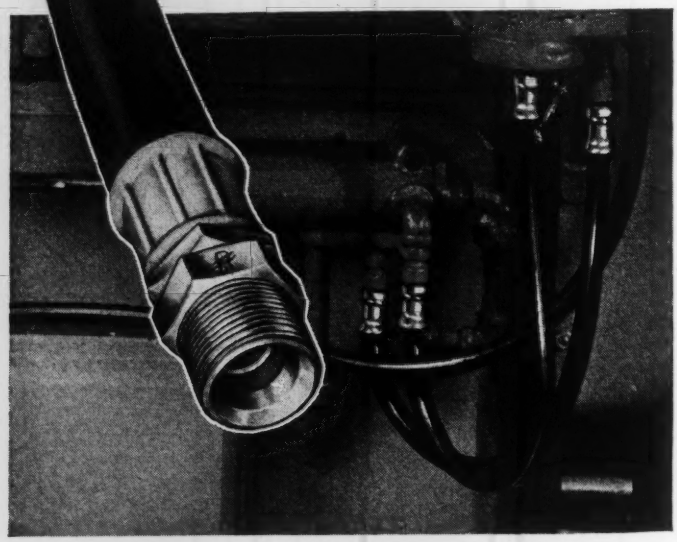

Resistoflex hose has what it takes for long service in hydraulic applications. Its capacity to endure constant flexing, vibration, shock loads—plus its remarkable inertness to oils—have proved to be the right answer for reduced troubles and replacements. That's why, for years, *Resistoflex* hose assemblies have been standard equipment in products of many well-known manufacturers.

With a high tensile compar tube, Resistoflex assemblies possess more than the necessary strength for medium as well as low pressure installations. Since this compar tube is so completely unaffected by oils, there's never any swelling, gumming or erosion—your guarantee of unclogged hydraulic cylinders and pump!

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CORPORATION

Belleville 9, New Jersey

SYNTHETIC FLEXIBLE PRODUCTS AND PARTS FOR INDUSTRY

Modern Machine

(*Concluded from Page* 111)

set up during operation
2. Movement of any member by clamping pressure preparatory to operation
3. Thermal expansion.

The first two sources posed no problem. Because of the relative massiveness of the machine members and the extremely light stresses during operation, elastic deflections are negligible. Use of the reed-type, noninfluencing clamp for locking worktable and a constant-pressure clamp for locking the main spindle housing to the column precluded clamping error.

Thermal expansion proved to be the most difficult condition to correct. Although atmospheric temperature can be controlled by air conditioning, difficulties arose from the many sources of temperature variation within the machine.

Temperature variation in the jig grinder resulted in two highly undesirable phenomena:

1. Change in the preload of ball bearings
2. Errors in workpiece resulting from expansion and contraction of location control members.

Location of the pneumatic cylinders on either side of the hydraulic cylinder in the yoke, in addition to avoiding a mechanical stress problem, provided an almost perfect temperature compensation condition. While fluid friction causes a slight rise in temperature in the hydraulic portion of the assembly, the cooling effect of expanding air in the adjacent air cylinders cancelled this and served to maintain a uniform operating temperature of about 70 F in this assembly.

Unfortunately, the twelve-degree rise resulting from rotation of the hollow main spindle in its preloaded ball bearings was immediately apparent in locational errors in the work, *Fig.* 11, as this heat caused an expansion of the main spindle housing from its locating face on the column.

This phase of the problem required a design modification in two respects. Instead of conventional iron castings, the main spindle housing is cast from a special 36 per cent nickel iron having a coefficient of thermal expansion approximately one-fourth that of gray iron. To further stabilize this member a heating element is located in this casting, adjacent to the bearings.

This heating element develops substantially the same Btu value as do the bearings, and is so connected to the motor control circuit that when the motor is running the heater is off, and vice versa; thus the housing is maintained at operating temperature at all times.

In the high-speed, vertical grinding spindles, temperature rise introduced the problem of variable ball-bearing preload. This was overcome by use of compression springs as a preloading means; thus, over the short range of dimensional change encountered, the axial load remained virtually constant.

WEIGHT REDUCED 20% APPEARANCE IMPROVED DEPENDABILITY ASSURED WITH STEEL CASTINGS

A peavy point is the business end of a peavy (named after the inventor) — a wooden handled tool used in logging.

Produced as a *fabrication,* the peavy point was made from ⅛" mild sheet steel rolled to shape and seam welded, with brackets welded to side on which to hinge a hook. A forged tip of high carbon steel was welded to tubular section, then drawn to a point and tempered.

By redesigning the peavy point to a one-piece *steel casting,* weight is cut 20%, permitting more efficient use of the peavy. Point is forged and hardened, same as with wrought steel tip . . . (this emphasizes the fact that *cast* steel — .40-.50 carbon steel in this case —can be forged and heat-treated to give results comparable to those obtained with rolled or otherwise fabricated steels.)

• • •

Here is another example of the engineering teamwork in design and redesign of parts which is resulting in lower costs and greater serviceability with steel castings.

This service is offered without cost or obligation. It makes available through your foundry engineer the full results of the development and research program carried on by Steel Founders' Society of America.

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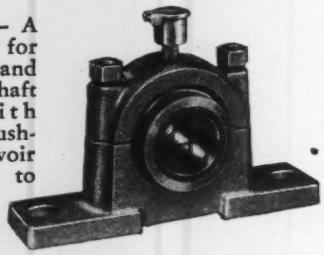
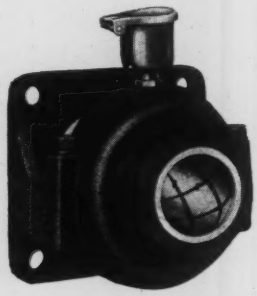
THERE'S A
RANDALL
PILLOW
BLOCK
FOR EVERY TYPE
APPLICATION

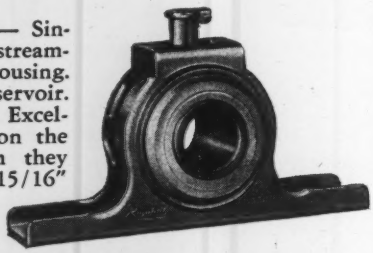
Normal Duty Pillow Block — A general service pillow block for normal duty on small shafts and for heavier duty on larger shaft sizes. Double-lubricated with graphited, phosphor bronze bushing in wool packed oil reservoir housing. For shafts 1¼" to 3 15/16" inclusive.





Flange or Side Mount Pillow Block—A top quality pillow block built for the most exacting service. Double-lubricated with graphited phosphor bronze bushing in wool packed oil reservoir housing. This pillow block is widely used in unit heaters, cabinet type air conditioning units. Side mounting only. For shafts ½" to 1 15/16" inclusive.

Sintered Bushing Series — Sintered bronze bushings in streamline one-piece steel housing. Wool packed oil reservoir. Mounts in any position. Excellent for general service on the smaller shafts for which they are available, ⅝", ¾", 15/16" and 1".



Write for catalog No. 49.—For additional information on these as well as other one piece or two piece pillow blocks available for light, normal or heavy duty. There's a Randall Pillow Block for every type application.

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DESIGN ABSTRACTS

(*Concluded from Page* 169)

proximately 85 per cent of the patents coming before them. While this slaughter was going on, the late President Roosevelt said "Patents are the key to Technology; Technology is the key to production." Quite recently, in a dissenting opinion, Mr. Justice Jackson, of the Supreme Court of the United States, remarked "The only patent that is valid is one which this court has not been able to get its hand on."

It appears that the courts have felt they must protect the public against temporary monopolies granted by the United States Patent Office. This attitude has persisted, even though the courts constantly uphold copyrights granted on songs, verses, cartoons, and the like, and in spite of the fact that the patent laws and the copyright laws find their source in the same paragraph of the Constitution.

Owners are now quite reluctant to sue on their patent rights for fear of having their patents invalidated. Powerful manufacturers can feel free to infringe patent rights of persons who do not have the money or the inclination to go into court and defend their rights, and especially when the possibility of success is small.

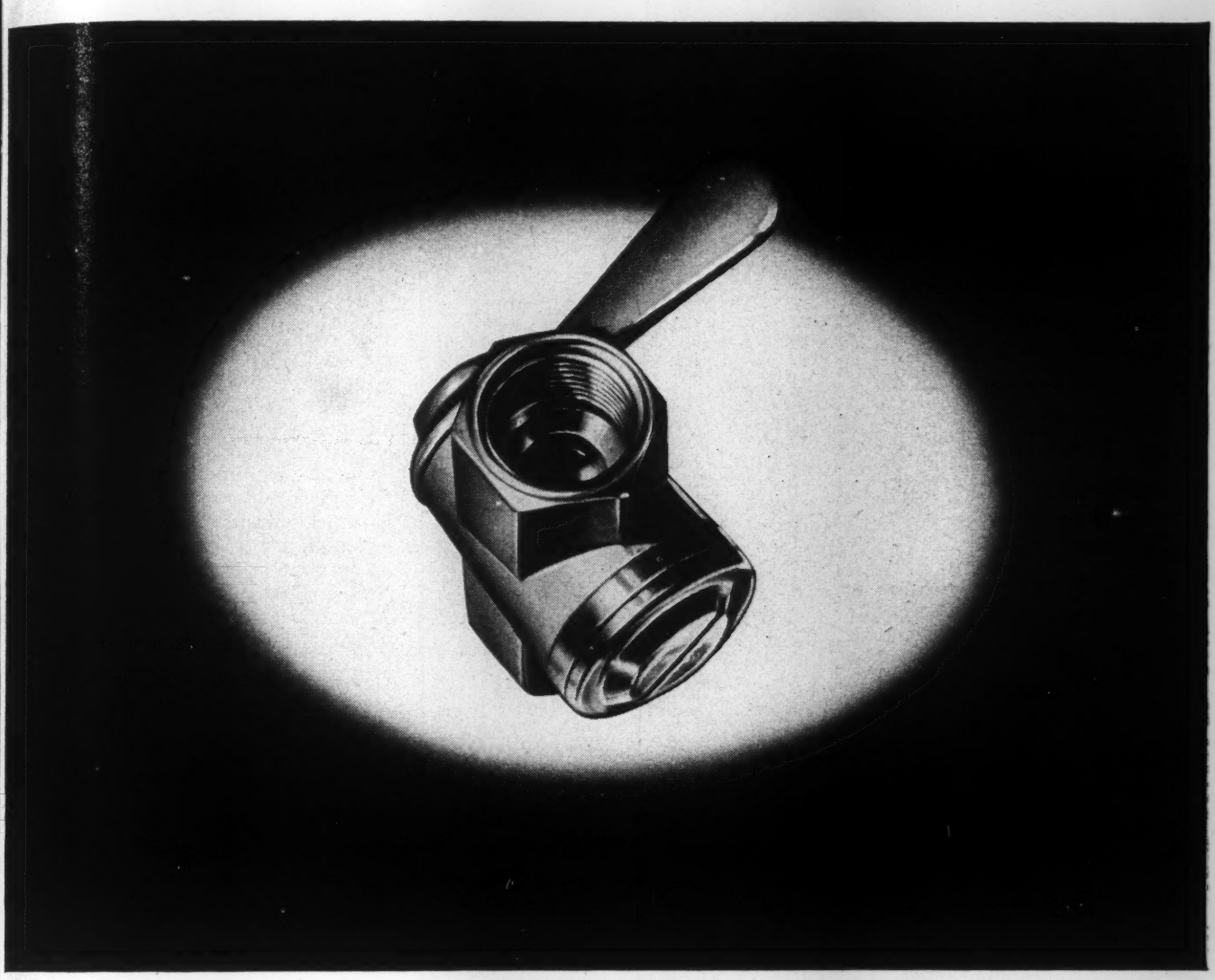
Changes Under Discussion

In recent years Congress has been deluged with bills proposing to revise the patent system. Many of these contain proposals which appear ridiculous to those acquainted with patents. Nevertheless, they represent a trend of thinking which cannot be ignored.

A bill which may possibly receive favorable consideration was recently introduced. Among other things, it provides that subject matter shall not be patentable if the differences between it and subject matter previously known would have been obvious at the time the invention was made to a person having ordinary skill in the art.

The patent system is a vital and a necessary part of our industrial life. In spite of all the ills attributed to it, its resulting effect is deemed to be good. At the present time the patent system needs clarification, if not revitalization. The issued patent should carry a very strong presumption of validity. At present it is an instrument which apparently can be invalidated by anyone who is willing to expend the money and energy. This requires better examination in the United States Patent Office, and a clear and consistent treatment of patents by the courts. Legislative action may possibly clarify the situation by enunciating a clear definition of what constitutes invention and how patents may be handled without violation of the antitrust laws or antimonopoly practices.

From a paper presented at the SAE National Aeronautic Meeting in Los Angeles, September 28-30, 1950.



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KOHLER PRECISION PARTS are used by manufacturers of Diesel engines, automobiles, trucks, tractors, jet engines, military, commercial and private aircraft, and for agricultural and industrial test equipment.

With an organization experienced and skilled in precision workmanship, Kohler maintains full facilities for forging, casting, machining, finishing, assembling, and testing.

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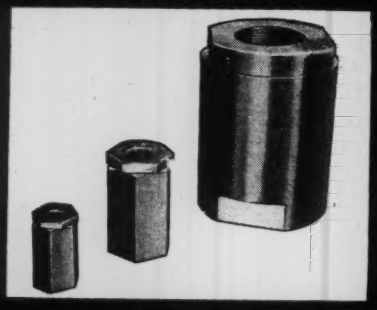
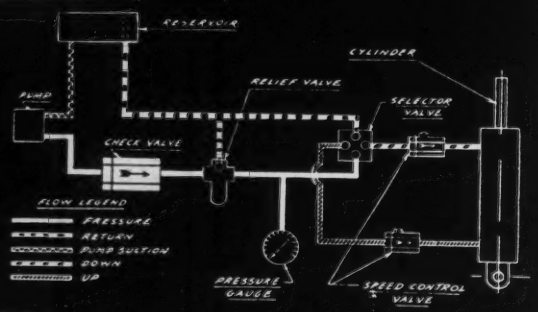
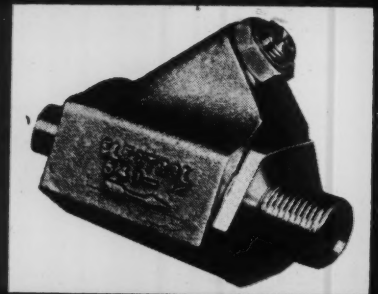
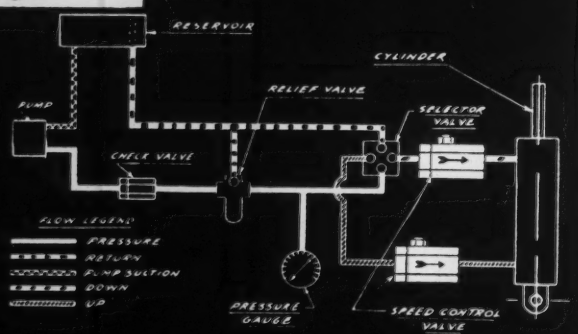




SPEED CONTROL VALVES





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FOR BETTER HYDRAULIC DEVICES

NEWS
OF MANUFACTURERS

A controlling stock interest in the Commonwealth Electric Co. Ltd., Welland, Ont., has been purchased by **The Reliance Electric and Engineering Co.**, Cleveland, O., manufacturer of industrial electric motors, generators and engineered drives. The manufacture of transformers and electric motors will be continued in the Welland plant. In addition, expansion plans anticipate the production there of Reliance V*S drives and a number of other specialized electric drives regularly supplied by Reliance to major industries in the United States and Canada.

♦

The **Parts Division of Sylvania Electric Products Inc.** has erected a new 30,000 sq ft plant in Warren, Pa. The structure will be devoted to the production of plastic and plastic-metal components for the radio, television and lighting industries. Flexibility of layout will permit an expansion to 100,000 sq ft in the future.

♦

The **Wm. K. Stamets Co.**, Pittsburgh, Pa., manufacturer of machine tools and special machinery, has purchased **The Enterprise Co.**, Columbiana, O., builders of special machinery. The Stamets firm will operate the Enterprise plant as a contract machine shop, and in addition, will build Stamets mill-type threading and finishing machinery there. Manufacture of the Enterprise line of sawmill machinery will be continued.

♦

The **Westinghouse Electric Corp.** will add a 100,000 sq ft wing to its lamp plant in Little Rock, Ark. Completion is scheduled for April, 1951.

♦

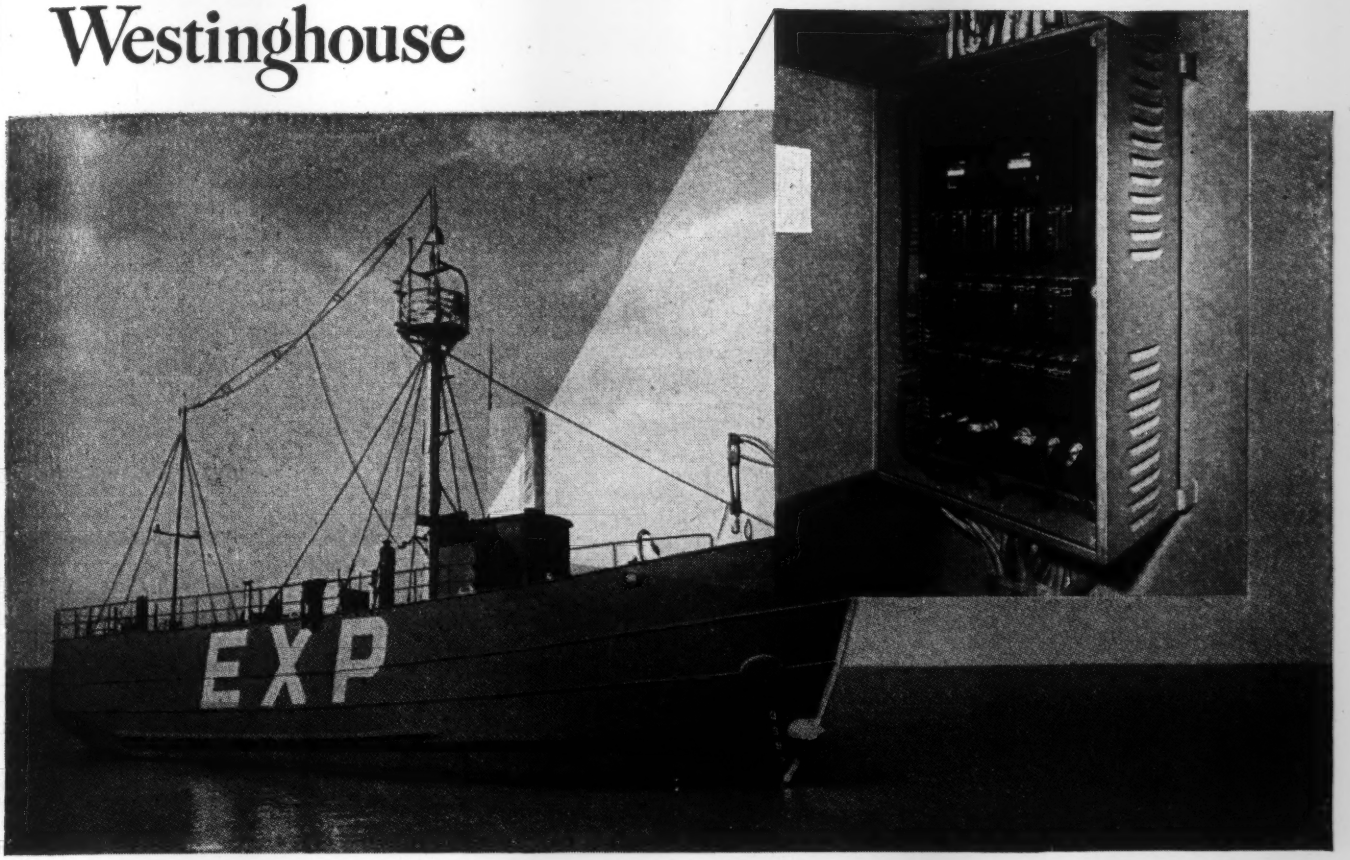
The **Minnesota Mining & Mfg. Co.**, as agents for the federal rubber reserve agency, and the **Pacific Rubber Co.**, as associates, have been awarded a contract to reactivate and operate a $22,000,000 government-owned synthetic rubber plant at Torrence, Calif.

♦

The **General Electric Co.** has formed a Mobilization Planning Committee within its Apparatus Department. Responsibilities of the new group involve the co-ordination of requests from the Department of Defense and

YOU CAN BE **SURE**.. IF IT'S

Westinghouse



AB BREAKERS GUARD VITAL CIRCUITS

on "remote control" lightship

Only an "Electronic" crew mans this U. S. Coast Guard robot lightship "EX P-99". Here, where circuits are remotely controlled automatic and trouble-free protection is essential. Westinghouse AB Breakers answer both requirements.

A harmless overload encountered in motor starting or initial surge of lights could trip an ordinary safety device . . . and might demand a time-wasting trip to the lightship. Not so, with the time-lag feature of AB Breakers. They "hold on" to harmless overloads—but trip instantly when actual protection is needed. Thus, service is uninterrupted because of harmless temporary overloads—yet positive, instantaneous protection is on THE JOB when it's really NEEDED.

AB Breakers can be grouped into a neat compact arrangement. The E and F frame units (mounted on the panelboard above) show how AB Breakers save valuable space... yet operating efficiency and convenience are never impaired! There are no fuses to replace, nothing to come apart or adjust, nothing that requires maintenance.

Westinghouse AB Breakers do the job with top performance, maximum safety and efficiency. Put them to work on your applications. Contact your nearby Westinghouse representative now for all the facts; or write for Bulletin DB 29-060, Westinghouse Electric Corporation, P. O. Box 868 Pittsburgh 30, Pa. J-30041

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Westinghouse

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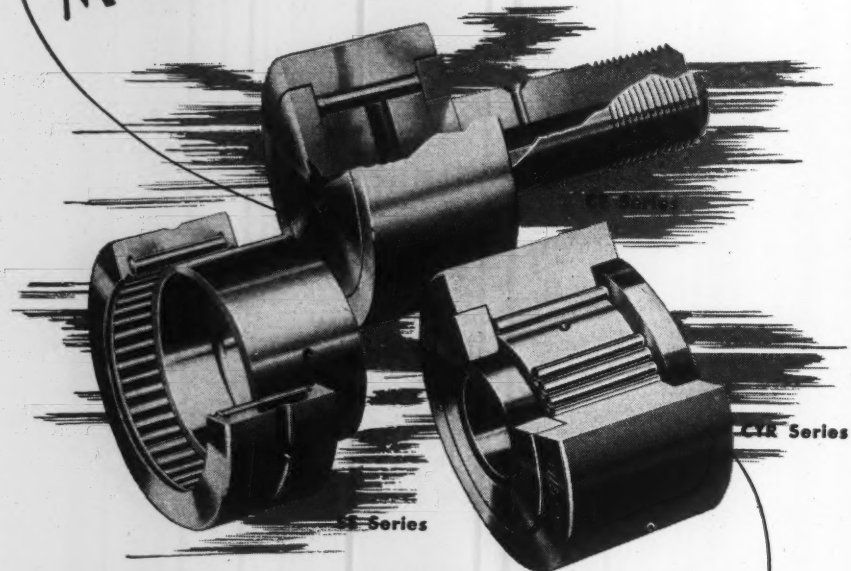
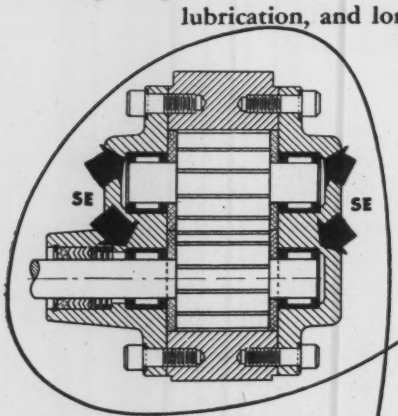
AB CIRCUIT BREAKERS

THE COMPLETE LINE

Costs less in the long run
McGILL® MULTIROL® Bearings

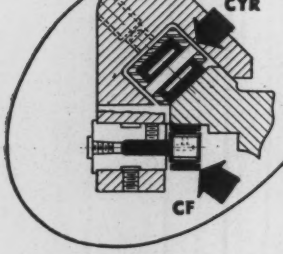


The bonus values found in McGill Multirol roller bearings are worth money to you. They mean increased profit measured in terms of savings . . . through improved anti-friction performance. This better performance is due to the consistently higher degree of precision . . . that marks all McGill bearings. For smooth performance, low starting torque, minimum maintenance, ease of assembly, simplified lubrication, and long life in your roller bearing applications . . . investigate the McGill SE, CF and CYR Multirol bearings.



SE Series in Gear Pump Application

Gear pump using MULTIROL SE Series Bearings replacing plain bearings on both drive and idler shafts for greater precision and longer life. Capacity is increased while radial bearing space is held to minimum of throat requirements.



CF and CYR Series in Slide and Carriage Application

MULTIROL Cam Followers and Cam Yoke Roller Bearings permits precision movement of Cross Slide and Carriage — eliminates costly machining and scraping of gibbs and ways. Simplifies assembly and lubrication. Also ideally suited to tail stocks.

McGILL® precision bearings

For informative literature write: McGill Manufacturing Co., Inc., Bearing Division, 200 N. Lafayette St., Valparaiso, Indiana.

its agencies for information concer ing facilities, personnel, possible pr duction schedules and other matte of material procurement planning f national defense.

♦

The Lapointe Machine Tool Co Hudson, Mass., has opened a new an enlarged plant of its British affiliat The Lapointe Machine Tool Co. Ltd Bushey, Hertfordshire, England. Re placing the former plant at nearb Edgware, the new works will be de voted exclusively to the manufac ture of broaching machines an broaches.

♦

A new company, **Ardec Inc.**, ha been formed with plant and genera offices at Media and South Orang St., Media, Pa. The corporation spe cializes in the production of preci sion parts and assemblies for the air craft, automotive, diesel engine, in strument, electronic, small arms and allied industries.

♦

Cummins Portable Tools, Chicago Ill., has purchased the assets of Fred W. Wappat Inc., Mayville, N. Y., manufacturer of portable electric hand saws for the industrial and heavy construction fields. The new prop erty will be operated by Cummins as the Fred W. Wappat Division. Wap pat personnel and identity of the Wappat line will be retained.

♦

SKF Industries Inc. is expanding productive facilities at its ball and roller plant in Philadelphia. The ad dition will provide more than 40,000 sq ft of floor space. Completion of construction is expected within six months; full use of the new facilities, however, is not expected until the middle of 1951.

♦

General Electric Co. has purchased the patents, name and technical data of the **Turbodyne Corp.**, Hawthorn, Calif., a **Northrup Aircraft Inc.** re search subsidiary. Turbodyne has re cently developed a turboprop air craft engine believed to be the most powerful propeller-type aircraft power plant.

♦

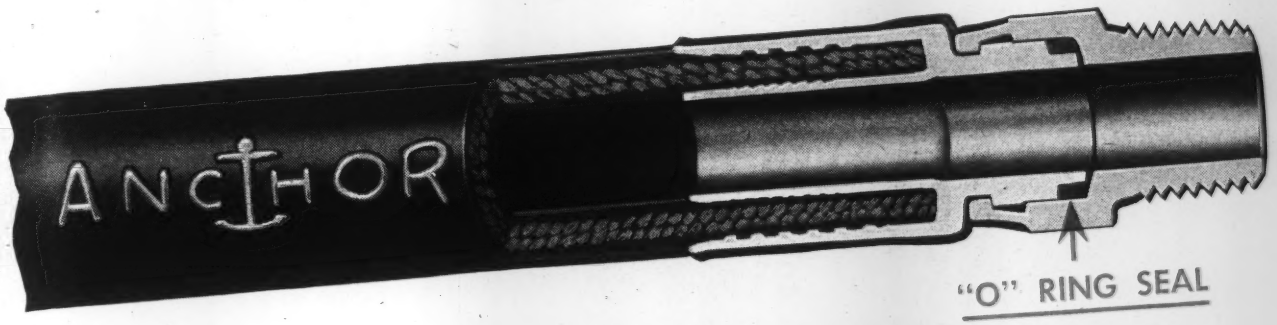
To more closely associate its name with its full scope of operations, Warner Electric Brake Mfg. Co., Be loit, Wis., has changed its name to **Warner Electric Brake & Clutch Co.**

♦

The **Marvel-Schebler Carburetor Division of Borg-Warner Corp.** is transferring its operations from Flint

SEALED FOR KEEPS!

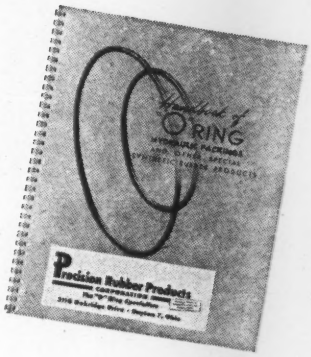
WITH THIS "O" RING SEAL



The swivel fitting on this hydraulic hose coupling manufactured by the ANCHOR COUPLING COMPANY INC., of Libertyville, Illinois, pioneers the use of an "O" Ring Seal for easy movement and positive sealing.

Use of this swivel male pipe thread coupling at one end of a hose assembly simplifies installation by eliminating one threaded joint and the usual pipe union. It avoids torsional strain because it permits the hose to assume its natural position when pressure is applied.

Since the coupling components are not only permanently swaged to the hose but also to one another, the sealing member must be 100% dependable right from the initial assembly. An "O" Ring from Precision Rubber Products Corporation proved to be the best packing obtainable for the leakproof and trouble free sealing.



PRP "O" RINGS
meet all
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● **Investigate PRP "O" Ring Seals today—send for your free copy of our "O" Ring Handbook**

It permits easy swiveling during hose installation assuring minimum wear and maximum seal life.

Space saving, leakproof "O" Ring Seals have solved countless design and production problems throughout industry. Write today to learn what PRP Engineers can do to help make yours a better product at lower cost.

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SPEED REDUCERS

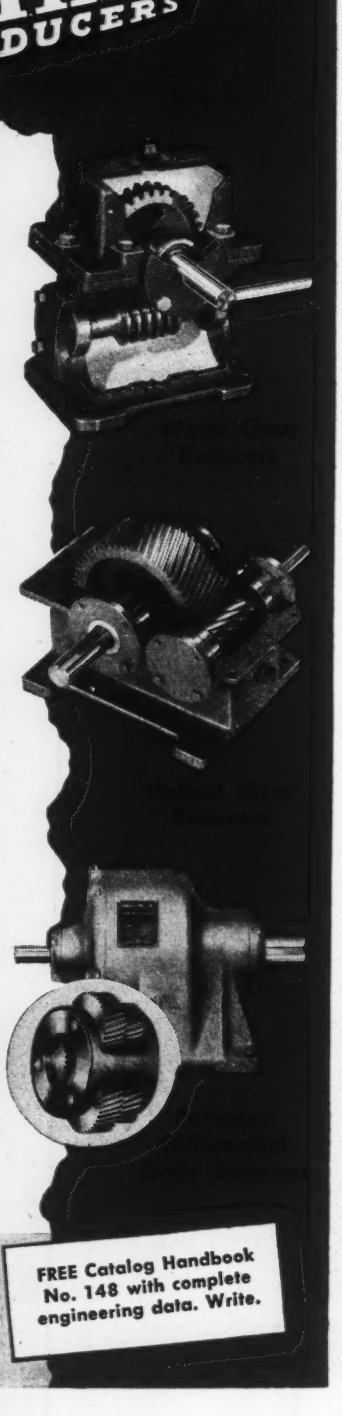
Management's penetrating interest in components of prime products goes deep . . . down to the last nut and bolt, as service reputation often hinges on individual parts, even those of outside manufacture. For the customer, failure of *any* part of the machine, regardless of cause or make, is a black mark against the *entire* product.

Vigilantly alert to this, the design engineer, the production manager, the purchasing agent, yes, even the sales manager—all concerned with the performance and marketability of primary equipment —are especially discerning in the selection of power transmission components.

Here you have one of the principal reasons that Winsmith Speed Reducers are so frequently selected . . . a preference based on enduring performance, the vital asset that means protection of a prime product's service reputation.

But you pay no premium for this protection. Actually, you save! For within the power range of fractional to 85 H.P., Winsmith has fully standardized on the widest selection of speed reducers in the industry . . . to serve you speedily and most economically.

Match *your* product's high reputation with components fabricated with the same care, backed by the same high reputation, that no link in your service chain may be weak. Winsmith power transmission engineers welcome the opportunity to consult. Write.



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16 ELTON STREET
SPRINGVILLE (Erie County), N. Y.

FREE Catalog Handbook No. 148 with complete engineering data. Write.

Mich., to Decatur, Ill., where it wi produce fuel pumps for jet engin planes in addition to its line of carbu reters. The company's name will b **Marvel-Schebler Products Division.**

♦

A new $1,000,000 plant for the manu facture of all types of ingot mold and accessories is being erected i Lansing, Ill., by the **Vulcan Mold an Iron Co.**, Latrobe, Pa. The new uni will produce approximately 50,00 tons annually, representing an in crease of over 50 per cent beyon Vulcan's present capacity.

♦

Pittsburgh Plate Glass Co. is pre paring the addition of a fourth tank with two drawing machines to it Clarksburg, W. Va., window glass producing plant. Cost of the struc ture and equipment is estimated at $500,000. The building program and installation are expected to reach completion by May, 1951.

♦

Construction of a new plant addi tion has been announced by the **Plas kon Division of Libbey-Owens-Ford Glass Co.** The plant addition will be erected on a site at Plaskon's prin cipal manufacturing location near Toledo, O.; the additional capacity will be in production by March, 1951.

♦

Gaston Power Tools, manufacturer of heavy-duty grinders and polishers for the metal working field and pro ducers of standby generators, has been sold to new interests. The new firm is known as Gaston Power Tools Inc. and will continue operations at 14829 Loomis St., Harvey, Ill.

♦

The **Charles Bruning Co. Inc.,** 100 Reade St., New York, manufacturer of blueprint and whiteprint papers and machines, drafting equipment, and technical photographic products, is planning the erection of a million-dollar plant at Bergen County Indus trial Terminal, Teterboro, N. J. Com prising 120,000 sq ft, the plant is ex pected to be in operation next year.

♦

B. F. Goodrich Co., through its division, B. F. Goodrich Chemical Co., has made arrangements to purchase from **American Home Products Co.** its subsidiary, **Harmon Color Works Inc.** The latter company is engaged in the manufacture of organic color pigments and dyes; operating plants are located in Kearny, N. J., and Haledon, N. J.

♦

The **Plastic and Rubber Products Co., Los Angeles, Calif.,** has been in corporated under the same name.

THE BIG TREND IS TO OSTUCO TUBING

For more strength, less weight, smooth modern beauty, and lower cost, product designers and manufacturers are turning more and more to steel tubing—and the big trend is to OSTUCO Tubing.

Ostuco's "Single Source Service" is the reason. By performing all three basic tubing operations—manufacturing, shaping, and fabricating—at one plant, Ostuco boosts quality and eliminates expensive, time-consuming shipments from one supplier or one location to another. Unified control of production assures precision in meeting your most exacting requirements; responsibility is clearly fixed.

Ostuco produces both seamless and electric welded steel tubing in a broad range of sizes and shapes . . . performs ALL types of forging and fabricating operations . . . supplies finished or semi-finished products. Write direct or to our nearest sales office for informative free booklet "Fabricating and Forging Steel Tubing."

From Your Blueprint . . . to Your Product

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Tapering • Swaging • Flanging • Bending
Upsetting • Expanding • Reducing • Beading
End closing • Spinning • Drilling • Slotting
Notching • Flattening • Shaping • Trimming
Threading • Angle Cutting • And Many Others

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Manufacturers and Fabricators of Seamless and Electric Welded Steel Tubing
Plant and General Offices: SHELBY 5, OHIO

SALES OFFICES: BIRMINGHAM, P. O. Box 2021 • CHICAGO, Civic Opera Bldg., 20 N. Wacker Dr. • CLEVELAND, 1328 Citizens Bldg. DAYTON, 511 Salem Ave. • DETROIT, 520 W. Eight Mile Road, Ferndale • HOUSTON, 6833 Avenue W, Central Park • LOS ANGELES, Suite 300-170 So. Beverly Drive, Beverly Hills • MOLINE, 617 15th St. • NEW YORK, 70 East 45th St. • PHILADELPHIA, 1613 Packard Bldg., 15th & Chestnut • PITTSBURGH, 1206 Pinewood Drive ST. LOUIS, 1230 North Main St. • SEATTLE, 3104 Smith Tower SYRACUSE, 501 Roberts Ave. • TULSA, 733 Kennedy Bldg. • WICHITA, 622 E. Third St. • **CANADIAN REPRESENTATIVE:** RAILWAY & POWER CORP., LTD., New Glasgow, Montreal, Noranda, North Bay, Toronto, Hamilton, Windsor, Winnipeg, Edmonton, Vancouver.

Here's why those in the know

–demand

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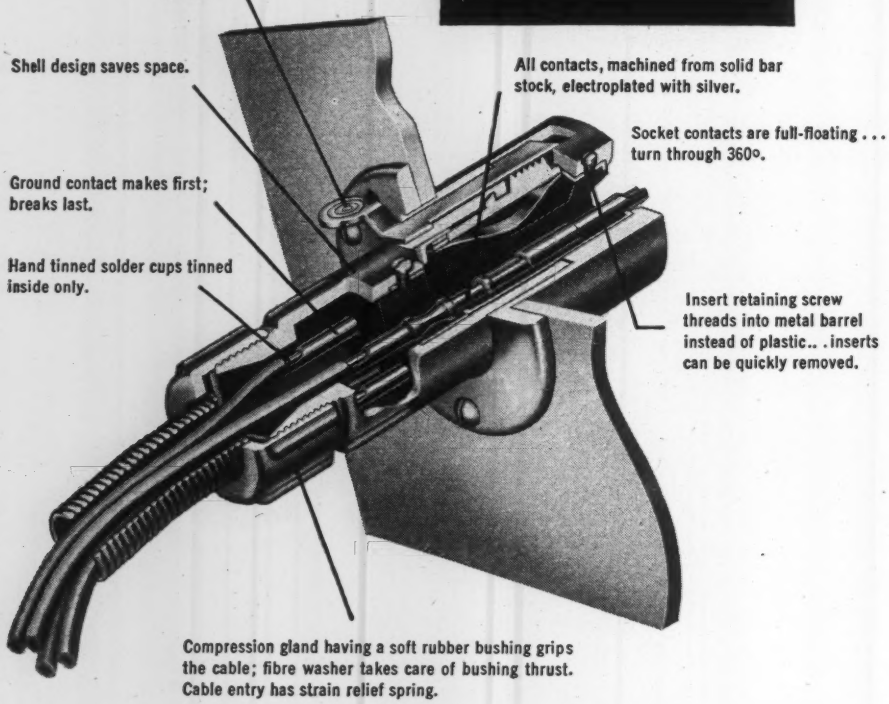
Patented exclusive latchlock device
Shell design saves space.
All contacts, machined from solid bar stock, electroplated with silver.
Socket contacts are full-floating . . . turn through 360°.
Ground contact makes first; breaks last.
Hand tinned solder cups tinned inside only.
Insert retaining screw threads into metal barrel instead of plastic.. . inserts can be quickly removed.
Compression gland having a soft rubber bushing grips the cable; fibre washer takes care of bushing thrust. Cable entry has strain relief spring.

No corners are cut . . . nothing is overlooked to assure you outstanding performance with Cannon Plugs. So long an engineer's choice, the words "Cannon Plugs" have become part of our electrical language. Continued excellence of design . . . ability to meet your changing requirements . . . are good reasons why the Cannon line of connectors continues to excel where specifications must be met. XL Connector Series is just one of the many Cannon types—world's most complete line. Request bulletins by required type or describe your needs.

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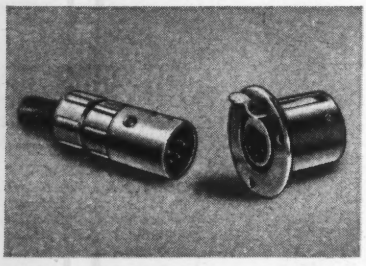


There are 12 items in the XL line. Insert arrangements available: 3 – 15 amp. contacts, 4 – 10 amp. contacts – working voltage 250 volts. Zinc and steel plugs with bright nickel finish are standard. Satin chrome finish also available on steel plugs.

MEETINGS AND EXPOSITIONS

Nov. 14-16—

American Institute of Electrical Engineers. Technical conference of Machine Tools Subcommittee on General Industry Applications to be held at the Sheraton Hotel, Worcester, Mass. H. H. Henline, 33 W. 39th St., New York 18, N. Y., is secretary.

Nov. 26-Dec. 1—

American Society of Mechanical Engineers. Annual meeting to be held at the Hotel Statler, New York. C. E. Davies, 29 W. 39th St., New York, N. Y., is secretary.

Nov. 27-Dec. 2—

Power Show. Nineteenth national exposition of power and mechanical engineering to be held in Grand Central Palace, New York. Additional information may be obtained from Charles F. Roth, President, International Exposition Co., Grand Central Palace, New York 17, N. Y.

Nov. 29-Dec. 1—

Society for Experimental Stress Analysis. Annual meeting and exhibit to be held at Hotel Warwick, New York. W. M. Murray, P. O. Box 168, Cambridge 39, Mass., is secretary.

Dec. 11-12—

American Management Association. Production conference to be held at the Drake Hotel, Chicago. Edward K. Moss, 330 W. 42nd St., New York 18, N. Y., is public relations director.

Jan. 15-18

Plant Maintenance Show to be held in the Public Auditorium, Cleveland. Additional information may be obtained from Clapp & Poliak Inc., 341 Madison Ave., New York 17, N. Y.

Jan. 22-26—

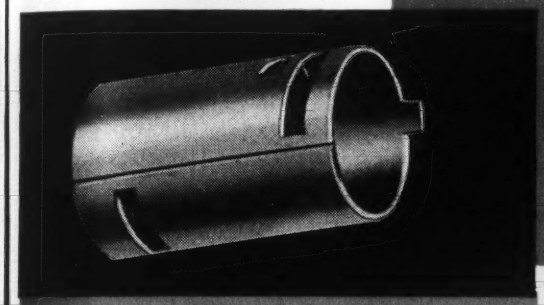
American Society of Heating and Ventilating Engineers. Tenth international heating and ventilating exposition to be held at the Commercial Museum in Philadelphia, Pa. Additional information may be obtained from Clapp & Poliak Inc., 341 Madison Ave., New York 17, N. Y.

rolled-type

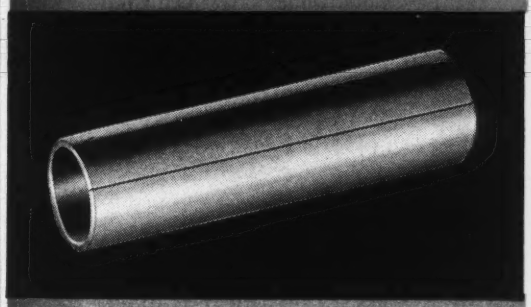
Bushings

from

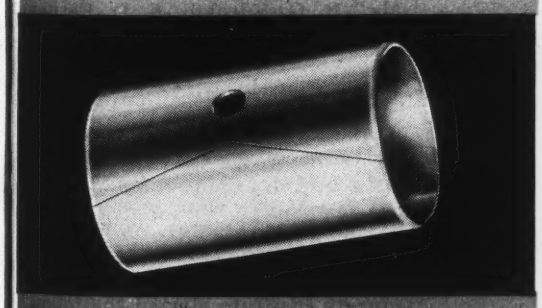
BI-METALLIC STRIP



BRONZE ON STEEL



BABBITT ON STEEL



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Performance at low cost . . .

The performance advantages of bi-metallic strip are now available in "Superformed" rolled-type split bushings and washers produced by our National Formetal division.

Bronze on steel, tin-base or lead-base babbitt on steel, and sintered copper-lead alloys on steel are available in a wide range of lengths and diameters.

They can be produced with ball-indented, oil-holding pockets on the lining side; with or without oil holes, grooves, notches, windows or cut-outs; and with straight, V or special seams.

A wide variety of washers, spacer tubes and ferrules is also produced in brass, steel or bronze.

Engineering counsel is available without obligation. Your inquiry is invited.

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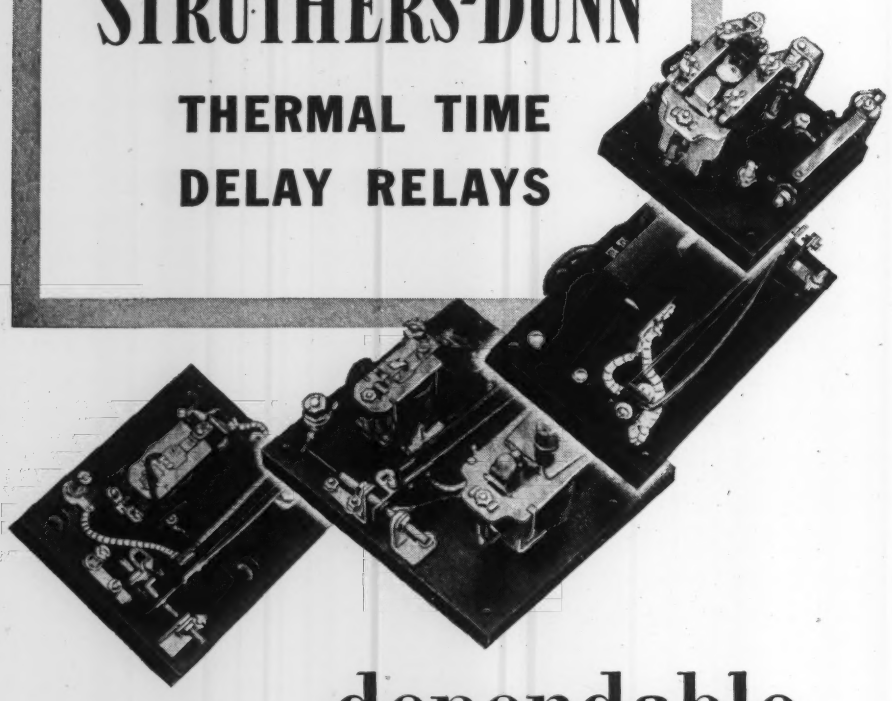
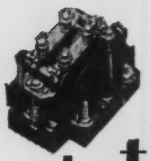
OVER FIFTY YEARS OF CONTINUOUS BEARING EXPERIENCE

STRUTHERS-DUNN

THERMAL TIME DELAY RELAYS





dependable ...low cost thermal timing

Struthers-Dunn thermal time delay relays offer a broad, economical selection for hundreds of applications where highly precise motor-operated timers represent unnecessary expense. Standard timing accuracy is ±10%. Suitable units are supplied for practically any specified timing from 1 second to a matter of minutes. Non-immediate recycling types that function through heating action only and immediate recycling types utilizing both the heating and cooling cycles are available.

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SOCIETY ACTIVITIES

The Machine Design Division of the **American Society of Mechanical Engineers** has tentatively scheduled the following sessions to be held during the Annual Meeting of ASME, Hotel Statler, New York, November 27—December 1.

Wednesday, Nov. 29, 9:30 am

"Some Design Aspects of Metal-Powder Parts," D. C. Bradley, New Jersey Zinc Co.

"The Fields of Utility of Investment Casting," Rawson L. Wood and D. Von Ludwig, Arwood Precision Casting Corp.

Wednesday, Nov. 29, 2:30 pm

"Furnace Brazing of Machine Parts," H. M. Webber, General Electric Co.

"Determination of the Dynamic Coefficient of Friction for Transient Conditions," G. C. Gould, Naval Ordnance Laboratory

Thursday, Nov. 30, 9:30 am

"Machine Design as a Career," J. F. Downie Smith, Iowa State College

"Dynamic Loading of Chain Drives," William K. Stamets Jr., consulting engineer

Thursday, Nov. 30, 2:30 pm

"Sealing with Rubber — Relation of Design to Properties of Elastomers," J. W. Swartz, Linear Inc.

"Studies of Relaxation Characteristics of Nonmetallic Gasket Materials," R. G. Farnam, F. D. Farnam Co.

"Rubber and Plastics in Packings," F. C. Thorn, Garlock Packing Co.

The Thursday afternoon session—a symposium on gaskets—will be held jointly with the Rubber and Plastics Division.

♦

The **Packaging Machinery Manufacturers Institute** elected the following officers at its Eighteenth Annual Meeting: President—Boyd H. Redner, general manager, Battle Creek Bread Wrapping Machine Co.; first vice president—Wilhelm B. Bronander Jr., vice president, Scandia Mfg. Co.; second vice president—G. Radcliffe Stevens, president, Elgin Mfg. Co.

♦

The Annual Meeting and Exhibit of the **Society for Experimental Stress Analysis** will be held at Hotel Warwick, New York, November 29—De-

cember 1. Commercial exhibits will show equipment emphasizing recent developments; educational exhibits will show interesting techniques, gadgets and tests. The exhibits will be open to all interested without charge on November 30 from 9:00 a.m. to 5:00 p.m. Technical sessions will run throughout the three-day meeting.

◆

At its 22nd Annual Meeting held in Cincinnati, O., the **Gray Iron Founders' Society Inc.** elected the following officers: President— E. L. Roth, Motor Castings Co.; vice president— R. G. Schaefer, (formerly secretary), Schaefer-Goodnow Foundries Inc.; treasurer—H. J. Trenkamp, (re-elected), Ohio Foundry Co.; secretary— J. S. Parrish Jr., Richmond Foundry & Mfg. Co.

For his outstanding contribution to the scientific advancement of the gray iron industry, Oliver Smalley, president, Meehanite Metals Corp., was presented with the Gold Medal Award, the society's highest honor.

◆

To assist manufacturing concerns with the organization of Standards, Dr. John Gaillard, mechanical engineer, **American Standards Association** and lecturer, Columbia University, will hold a private five-day seminar January 22-26, 1951, in the Engineering Societies Bldg., New York. The seminar, designed specifically for company representatives, will consist of ten conferences, each comprising a lecture by Dr. Gaillard and a round-table discussion. Details may be obtained by writing to Dr. Gaillard, 400 West 118 St., New York 27, N. Y.

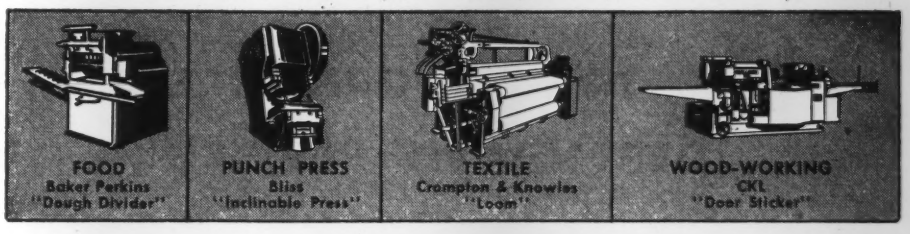
◆

Howard C. Wolf, assistant to the president of Mullins Mfg. Co., was elected president of the **Pressed Metal Institute** at its annual convention at Tadoussac, Quebec. Other officers named are: Vice president of PMI—Thomas L. Baker, vice president of the National Stamping Co.; and secretary-treasurer — Hunter Morrison Jr., vice president of Morrison Products Inc.

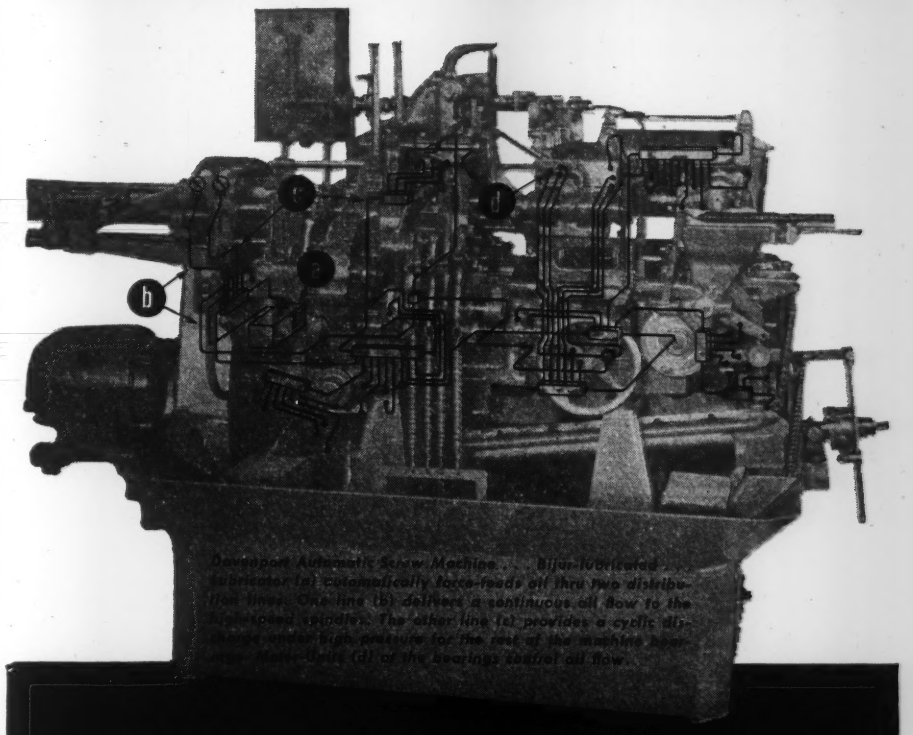
◆

At its 51st Annual Convention in New York, the **National Metal Trades Association** elected the following officers: President—Joseph L. Kopf, president, Jabez Burns & Sons Inc.; first vice president — Philip M. Morgan, president, Morgan Construction Co.; second vice president and treasurer — Charles S. Craigmile, president, Belden Mfg. Co.

LEADING MANUFACTURERS BUILD IN THE BIJUR SYSTEM



automatic lubrication



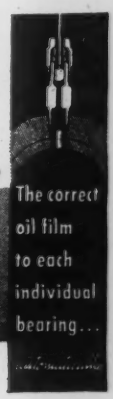
boosts production and cuts maintenance

Designing an automatic lubricating system into a machine doubles your prospects for user satisfaction.

First, maintenance time is converted into more time for production. A Bijur-lubricated machine oils itself during operation, doing away with wasteful down-time for lubrication.

Next, maintenance costs are cut as effective bearing life is lengthened. The Bijur system provides positive Meter-Unit control of oil flow *at the bearings*, considerably reducing bearing wear.

When you design that new machine, call on the Bijur engineer. Meanwhile, for further information write for "The Travels of Modern Lubrication."



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LUBRICATING CORPORATION

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SEALMASTER
BALL BEARING UNITS
For Improved Product Performance

Design engineers find that SealMaster Units have *all* the essential bearing features which assure high-efficiency, smooth-running new or rebuilt equipment. SEALMASTERS are used to improve the performance of such diversified products as conveyors, air conditioning equipment, heating installations, textile machinery, machine tools, farm equipment and many other types of machines and equipment.

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These Features Pay Off!

1 PERMANENTLY SEALED!
Felt-lined steel flinger rotating in labyrinth prevents entry of dirt or loss of grease.

2 SELF-ALIGNING!
Bearing unit, with seals independent of housing, aligns itself without seal distortion.

3 PRE-LUBRICATED!
Proper amount of lubricant is placed in bearing chamber before shipment . . . Unit is ready for immediate use.

4 NO HOUSING WEAR!
Patented locking pin and dimple prevent rotation of outer race in housing, and positions bearing for re-lubrication . . . *No housing wear means quiet operation.*

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STEPHEN S-A ADAMSON MFG. CO.
18 Ridgeway Avenue, Aurora, Illinois — Los Angeles, Calif. • Belleville, Ontario

Factory Representatives and Dealers in All Principal Cities

SALES AND SERVICE PERSONNEL

TO DIRECT the marketing and sales of Shakeproof precision fastenings, **John S. Hawley** has been appointed marketing manager of Shakeproof Inc., division of Illinois Tool Works. Before joining Shakeproof Mr. Hawley was sales manager of Fanner Mfg. Co., Cleveland. Prior to this he was associated with the Carborundum Co., Niagara Falls, N. Y. and for fifteen years with appliance divisions of General Electric Co.

◆

Appointment of **P. L. Edwards** as manager of its central district office in Pittsburgh has been announced by Raybestos-Manhattan Inc., Manhattan Rubber division, Passaic, N. J. He succeeds **R. C. Rice**, who has retired from active service with the company.

◆

Kennametal Inc., Latrobe, Pa., recently announced the following appointments: **William Bruun** has been named representative in the Midwestern district, working out of the Chicago office, 5830 West 26th St., and **F. O. Hill** has been appointed service engineer in the Cincinnati district, working out of the Cincinnati office at 2162 Gilbert Ave. Also, the Kennametal district office address in Indianapolis has been changed to Room 620, Board of Trade Bldg., 143 North Meridian St., Indianapolis 4, Ind.

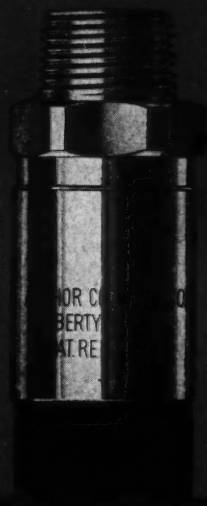
◆

Associated with the United States Rubber Co. for twenty-seven years, **Thomas H. Fitzgerald** has been named sales manager of reclaimed rubber for the Naugatuck chemical division. Other recent appointments include that of **Henry A. Rome** as manager of special products sales and **William C. Hall** as manager of molded goods sales.

◆

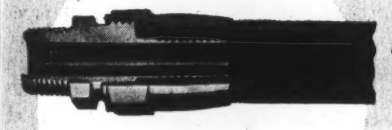
Frank B. Rackley, formerly executive vice president of the Jessop Steel Co., Washington, Pa., was recently elected president of the company. Mr. Rackley joined the company in June, 1948 as general manager of sales and was made vice president in charge of sales in October of the same year. Prior to joining the Jessop Steel Co., Mr. Rackley was manager of stainless sales for the western area of

Stop High-Pressure Leaks and Blow-offs



—with ANCHOR Ductile Sleeve Hose Couplings

Clamp-type Couplings

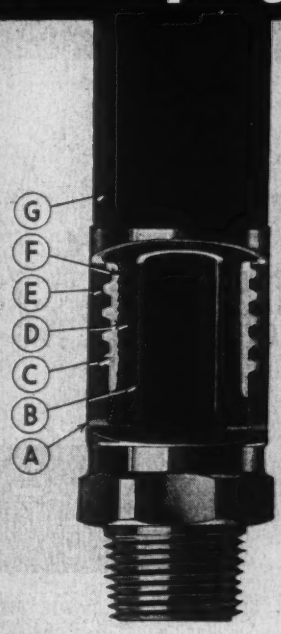


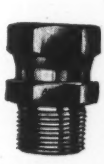
2-Piece Re-usable Couplings

Factory application of Anchor's exclusive Ductile Sleeve Coupling is your best insurance against leaks or blow-offs. Our superior grip gives you a plus margin of safety and a minimum time loss due to lay-up of equipment.

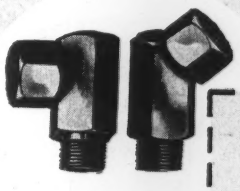
Here's how it works (referring to the illustration): Cover stock (G) is removed from the hose by grinding and buffing. Ductile Sleeve (C) is slipped on the end of the hose over the bare wire. Coupling body (A) with steel insert (B) attached is placed in position on the hose. The coupling is then swaged radially on the hose, imbedding the ductile sleeve into the internal grooves of the coupling shell (E) and into the mesh of the wire braid reinforcement (F). This is the exclusive Anchor patented grip.

Some of the applications of Anchor high-pressure flexible oil lines include high-pressure riveting, road machinery, snow plows, coal mining machinery, material handling machinery, machine tool applications, agriculture, railroads, oil field machinery, liquefied petroleum gas, lubrication and various other installations.





NPTF Straight Adapter Unions



45°-90° Adapter Unions

Other styles of related fittings available for piping installations

ANCHOR COUPLING CO. INC.

Factory: Libertyville, Illinois • Branch: Detroit, Michigan

Send for engineered information on Anchor high-pressure couplings for your particular job requirements.

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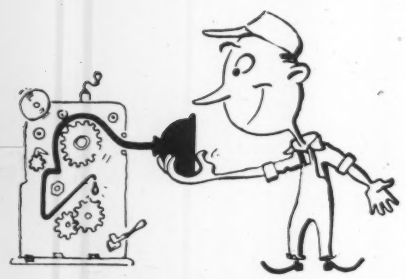
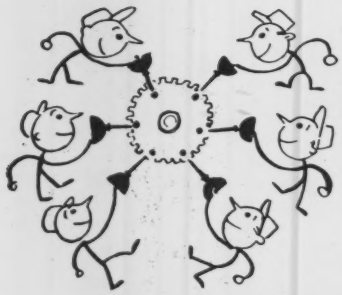
Manzel AUTOMATIC FORCE FEED LUBRICATION

Meters the precise amount of oil needed.

Reaches vital parts ordinary methods can't lubricate.





Gives each wearing point a full-time 'oiler'.

Frees manpower for more productive jobs.

Manufacturers using Manzel Lubricators report that they save their initial cost many times over in reduced labor cost, lower oil consumption, and fewer breakdowns. "Manzels" are standard equipment on many makes of engines, pumps, compressors, hydraulic presses, conveyors, and other machinery. Or...you can install them on present equipment.

Manzel representatives will gladly supply technical assistance on lubrication problems.

Manzel

Division of FRONTIER INDUSTRIES INC.

276 BABCOCK ST., BUFFALO, N. Y.

the Carnegie Illinois Steel Corp. The company has also announced the appointment of **John R. Harbaugh** as assistant general manager of sales. He has been associated with the company for ten years, doing special assignment work in metallurgical sales. At the same time **Eugene N. Lohre** was named district manager of the Chicago territory with offices at 322 South Michigan Ave., Chicago, Ill., and **Homer A. Mawhinney** was appointed district manager of the southern territory with offices at 1760-F Valley Ave., Birmingham, Ala.

♦

E. C. Hanks was recently appointed sales manager for the gearing division of Westinghouse Electric Corp., Pittsburgh. A Westinghouse employee for about twenty years, Mr. Hanks comes to his present position from the company's Chicago offices, where he was gearing division representative for the Northwestern district.

♦

Identified with the sale of industrial supplies for a number of years, **George E. Wells** has been appointed representative for the Bristol Co. of Waterbury, Conn., in the states of Massachusetts, New Hampshire, Maine and Vermont. He will cover the company's complete line of socket screw products.

♦

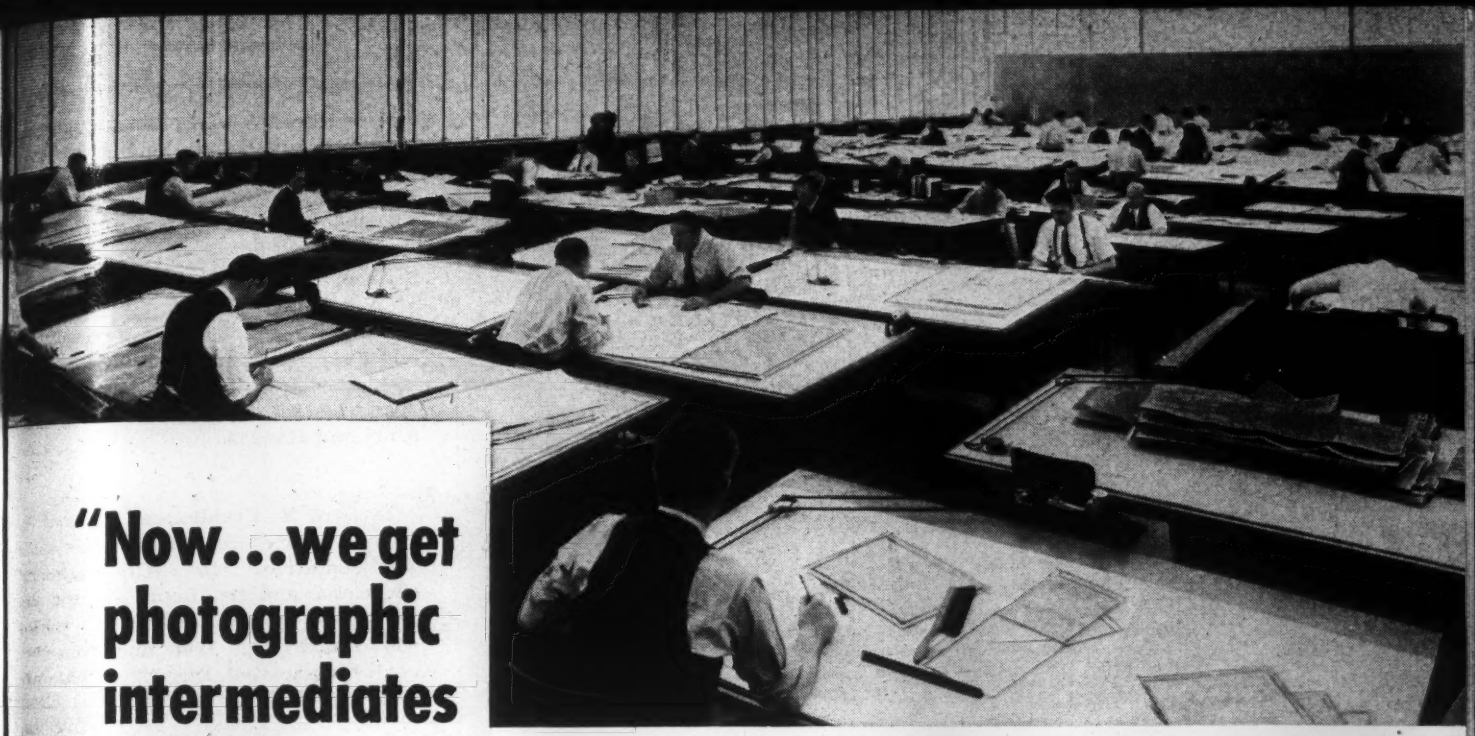
Edward A. Green has been appointed staff assistant to **A. W. Bartling**, general sales manager of General Electric Company's small apparatus divisions, with headquarters at Lynn, Mass. He will be replaced in his former position as manager of sales of the company's control divisions by **L. Dean Fowler,** who will now have full sales responsibility for all products of the control divisions. The company has also recently announced the appointment of **William R. Boggess** to assistant manager of sales of the fractional horsepower motor divisions at Fort Wayne, Ind.

♦

Appointment of **D. C. Wedlick** as district sales engineer in the Muncie-Anderson, Ind., area has been announced by the Michigan Tool Co. Recently service engineer with the company in Detroit, Mr. Wedlick joined Michigan Tool 15 years ago. His new headquarters will be at 401 Willow Drive in Muncie.

♦

Three new representatives have been appointed by Ajax Flexible Coupling Co. Inc. The Samson Sales Co., operated by **Elmer D. Samson**, 420 Market St., San Francisco, Calif.,



A section of the engineering department—The Heald Machine Company, Worcester, Mass.

"Now...we get photographic intermediates quickly, economically"

reports

THE HEALD MACHINE COMPANY

Since 1947 the production of photographic intermediates has been easier, faster, more economical for The Heald Machine Co. *Thanks to Kodagraph Autopositive Paper.*

With this revolutionary, low-cost intermediate material, positive photographic copies of original drawings are produced *directly*—without a negative step.

Furthermore, Heald uses its direct-process machine for the exposure operation; standard photographic solutions for development. And there's this convenience, too: darkroom handling is a thing of the past! Both operations are performed under ordinary room light.

How "Autopositive" is used: A large percentage of the precision finishing machines manufactured by Heald are specially engineered to customer-specifications. And, frequently, Heald is requested to draw special units and fixtures on the customer's own drafting paper. These originals are reproduced on "Autopositive" (for Heald's record) and turned over to the customer when the job is completed.

Thus, when repair parts are requested or a duplicate order is entered, shop prints can be made promptly from the Autopositives. These new intermediates have dense photographic black lines on an evenly translucent base . . . will not smudge or smear in print production . . . turn out sharp, clean prints at uniform, practical machine speeds.

Kodagraph Autopositive Paper

"THE BIG NEW PLUS" in engineering drawing reproduction

- **It enables you, or your blueprinter, to produce positive photographic intermediates *directly* . . . at a new low cost.**
- **It preserves valuable original drawings . . . cuts redrafting costs.**
- **It restores old, soiled drawings . . . gives you cleaner, sharper prints.**
- **It gives you "photo-lasting" file copies, which will not curl, turn yellow, or otherwise deteriorate.**

A new illustrated booklet, "Modern Document and Drawing Reproduction," gives all the facts on this revolutionary photographic intermediate material. *It's free. Just mail the coupon.*

EASTMAN KODAK COMPANY
Industrial Photographic Division
Rochester 4, N. Y.

19

Please mail a free copy of "Modern Document and Drawing Reproduction"—your new booklet on Kodagraph Autopositive Paper.

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TRADE-MARK

Uniform

STRENGTH, WEIGHT DUCTILITY, MACHINABILITY, WELDABILITY, I.D. and O.D.

Can be Bent,

FLANGED, EXPANDED, TAPERED, DEPRESS BEADED, EXPAND BEADED, ROLLED, EXTERNAL UPSET, INTERNAL UPSET, SPUN CLOSED, FORGED, BEVEL FLANGED, FLATTENED, SWAGED, FLUTED.

Consult us for engineering and technical help in the selection of tubing best suited to your needs.

Design simplification, fabricability, uniformity, ductility, ease of assembly and installation are just a few of the words we use to describe the many advantages of using MICHIGAN TUBING. Whether you manufacture automobile parts, power shovels, toys or typewriters, MICHIGAN TUBING can give you a better product at lower cost.

A Quality Product,

can be worked in your plant or prefabricated by MICHIGAN.

RESISTANCE WELDED STEEL TUBING

Michigan STEEL TUBE PRODUCTS CO.

THE OLDEST NAME IN ELECTRIC

33 Years in the Business

9450 BUFFALO STREET • DETROIT 12, MICHIGAN

FACTORIES: DETROIT, MICHIGAN • SHELBY, OHIO

DISTRIBUTORS: Steel Sales Corp., Detroit, Chicago, St. Louis, Milwaukee, Indianapolis and Minneapolis—Miller Steel Co., Inc., Hillside, N. J.—C. L. Hyland, Dayton, Ohio—Dirks & Company, Portland, Oregon—James J. Shannon, Milton, Mass.—Service Steel Co., Los Angeles, Calif.—American Tubular & Steel Products Co., Pittsburgh, Pa.—Strong, Carlisle & Hammond Co., Cleveland, Ohio—A. J. Fitzgibbons Co., Buffalo, N.Y.

will represent Ajax flexible couplings, Ajax Shaler-Shakers and Ajax Lo-Veyors in northern California. **J. S. (Steve) Thurlow,** manager of the Thurlow Equipment division of G. Donald Bradley Co., 4402 White Bldg., Seattle, Wash., will cover the states of Washington, Oregon and Idaho on all products except flexible couplings, and **Boyd Goodhart,** 1425 Troost Ave., Kansas City, Mo., will handle the complete Ajax line in western Missouri and Kansas.

♦

Richard M. Friddle and **Richard A. Frazee** have been assigned as sales representatives to Allis-chalmers' Cleveland and St. Louis district offices, respectively. Both men joined the company in 1948 and have recently completed the Allis-Chalmers graduate training course.

♦

In order to offer expanded sales and service facilities to the increasing numbers of users of phosphor bronze, nickel silver, cupro nickel, and beryllium copper, the Riverside Metal Co. of Riverside, N. J., has assigned **Richard E. Pilling** to assist **F. M. Shelley Jr.,** general sales manager at the Newark offices.

♦

Henry M. Sossaman was recently appointed general sales manager of Quaker Rubber Corp., division of H. K. Porter Co. Inc., Philadelphia. Mr. Sossaman's appointment to his new position culminates more than twenty-two years of service with the company. In his new capacity he will be in charge of sales of all products manufactured by Quaker Rubber Corp., including conveyor and transmission belting, hose, packing and miscellaneous industrial rubber products for all industrial uses.

♦

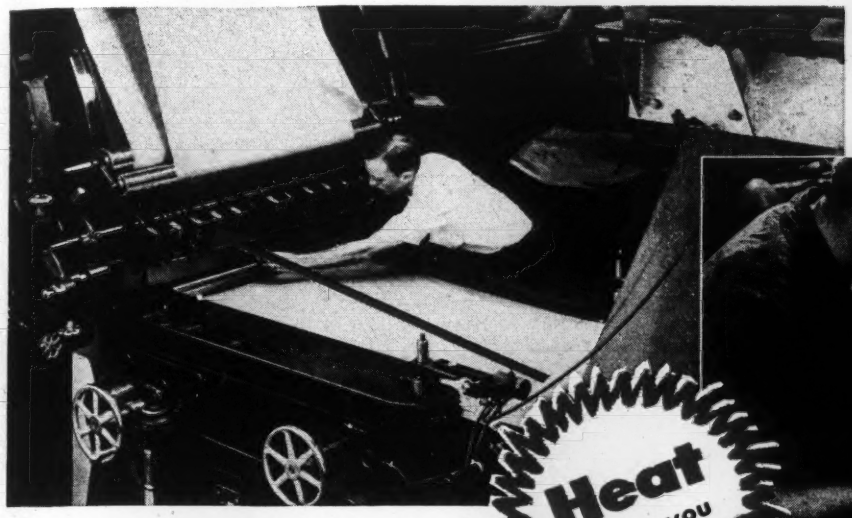
Appointment of **David R. Meslang** as manager of the Seattle branch of the Crucible Steel Company of America was announced recently. Mr. Meslang has been associated with the company since 1935, when he joined the San Francisco branch.

♦

Carboloy Co. Inc., Detroit, has announced the appointments of **J. E. Weldy** as marketing manager, **J. S. Gillespie** as product sales manager and **J. M. Bertotti** as field sales manager. Mr. Weldy will be responsible for all sales, merchandising and other marketing activities for both carbides and special metals. In his new capacity Mr. Gillespie will be directly responsible for technical selling operations on dies, tools, wear parts and

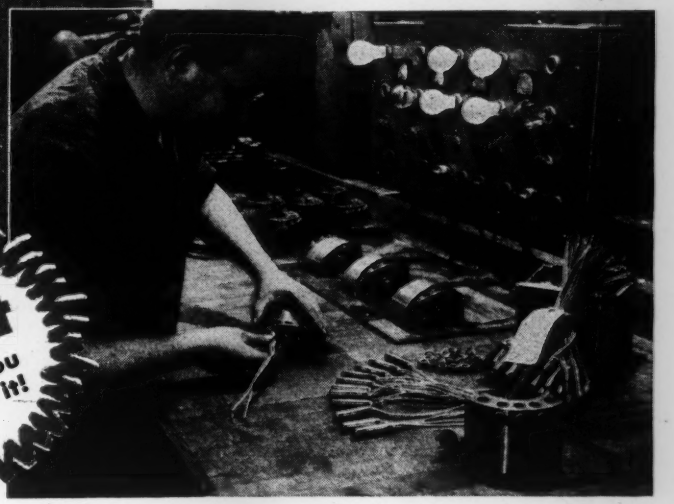
CALROD* HEATERS

"Simplest, easiest, and most reliable way to build heat into your product," says Morrison Machine Co.



This Morrison machine, built with G-E Calrod cartridge heaters, accurately shrinks the fabric in warp and weft. Temperatures, controlled by thermostats, are varied according to the fabric being shrunk.

Heat where you want it!



Only a few minutes are required to insert the cartridge heaters into the heating shoes, connect the leads, and fit the shoes over the blanketed cylinders of the shrinking machine.

LIKE THE MORRISON MACHINE CO., designers and builders of textile machinery, you may find that the simplest, easiest, and most reliable way to build heat into your product is with G-E Calrod heaters—tubular, cartridge, strip or fin.

The Morrison Co., faced with the problem of building intense, controllable heat into machines for shrinking textiles as they roll at high speed off the looms, picked G-E Calrod heaters. Here's why—

1. Use of G-E Calrod heaters cut machine manufacturing costs—workmen need only insert the heaters in the heating shoes and connect the leads.

2. Use of G-E Calrod heaters increases customer acceptance and satisfaction because the heaters are reliable and easy to operate.

The trademark, "Calrod," famous for 24 years, is a guarantee to customers that these heaters are designed to stand up under 'round-the-clock schedules. Customers can get maximum use of machinery and maximum output.

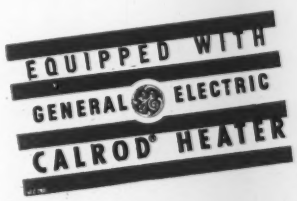
The heaters are easily controlled by thermostats, which can be set for the desired temperature up to 750 F.

** Registered Trademark of General Electric Co.*

Before You Design your equipment for repetitive manufacture, get together with the G-E heating specialist in your nearest G-E office. He'll be glad to help solve your heating problems by selecting from our stock of heaters available in a wide range of standard sizes, ratings, and sheath materials.

NEW, FREE CATALOG

Write us on your letterhead for Bulletin GEC-1005A. And, if you'll also tell us about your heating problems, we'll send you application bulletins to help you select and correctly apply the right heaters to give you heat *where* you want it . . . *when* you want it . . . and in the *amount* you want it. Section 720-29, Apparatus Dept., General Electric Co., Schenectady 5, N. Y.



NEW SELLING AID

These labels, with a gummed back to enable you to place them on your machinery, are now available.

GENERAL ELECTRIC

720-29

SHENANGO-PENN *Centrifugal* CASTINGS ... KEY TO SAVINGS

Bronze cylinder centrifugally cast and precision finished to specification by Shenango-Penn for use in manufacture of corrugated paper.

Major parts last longer ... cost less!

TAKE this special bronze forming roll cover, for example. Because it is a Shenango-Penn *centrifugal* casting, it combines complete uniformity, exceptionally fine, pressure-dense grain, and freedom from blow holes, sand inclusions and other common, often hidden defects.

The money-saving result, quite naturally, is greatly prolonged wear life and the plus strength that will take abnormal loads and speeds without risking costly failure in service.

This is just one of the countless types of cylinders, sleeves, liners, rings, rolls and roll assemblies produced by Shenango-Penn in ferrous and non-ferrous metals for various industries. When *you* need such parts . . . rough, semi-machined or precision finished, small or large . . . specify Shenango-Penn and profit by Shenango-Penn's specialized processes, facilities and experience.

MEANWHILE, ASK FOR BULLETIN

No. 143—Centrifugal castings of non-ferrous metals and alloys.

No. 144—Centrifugal and static castings of plain and alloyed irons.

No. 145—Centrifugally cast bushing stock in all standard sizes.

SHENANGO-PENN MOLD COMPANY
1282 West Third Street Dover, Ohio
Executive Offices: Pittsburgh, Pa.

SHENANGO PENN

All Bronzes • Monel Metal • Ni-Resist • Meehanite Metal®

mining equipment, and Mr. Bertotti will be in charge of Carboloy's district sales organization, distributor and jobber sales activities, and the company's customer training program.

♦

At a recent meeting of the board of directors, Admiral **Jonas H. Ingram,** U. S. N. (retired), was elected vice president of Reynolds Metals Co. His assignment will be that of vice president in charge of executive sales. At the same time it was announced that **John E. Blomquist** has been appointed industry manager of the company's industrial parts division.

♦

Merle E. Wendt, in charge of vinyl production development service and technical sales service on adhesives and latices in the chemical development laboratories of the Goodyear Tire & Rubber Co., has been promoted to assistant manager of the chemical products development division. Replacing Mr. Wendt is **Kermith K. Fligor,** long experienced in vinyl plastic development work. Goodyear Tire & Rubber Co. has also announced the transfer of **E. J. Dunlavy** of the vinyl film sales department from Akron to the New York offices of the company.

♦

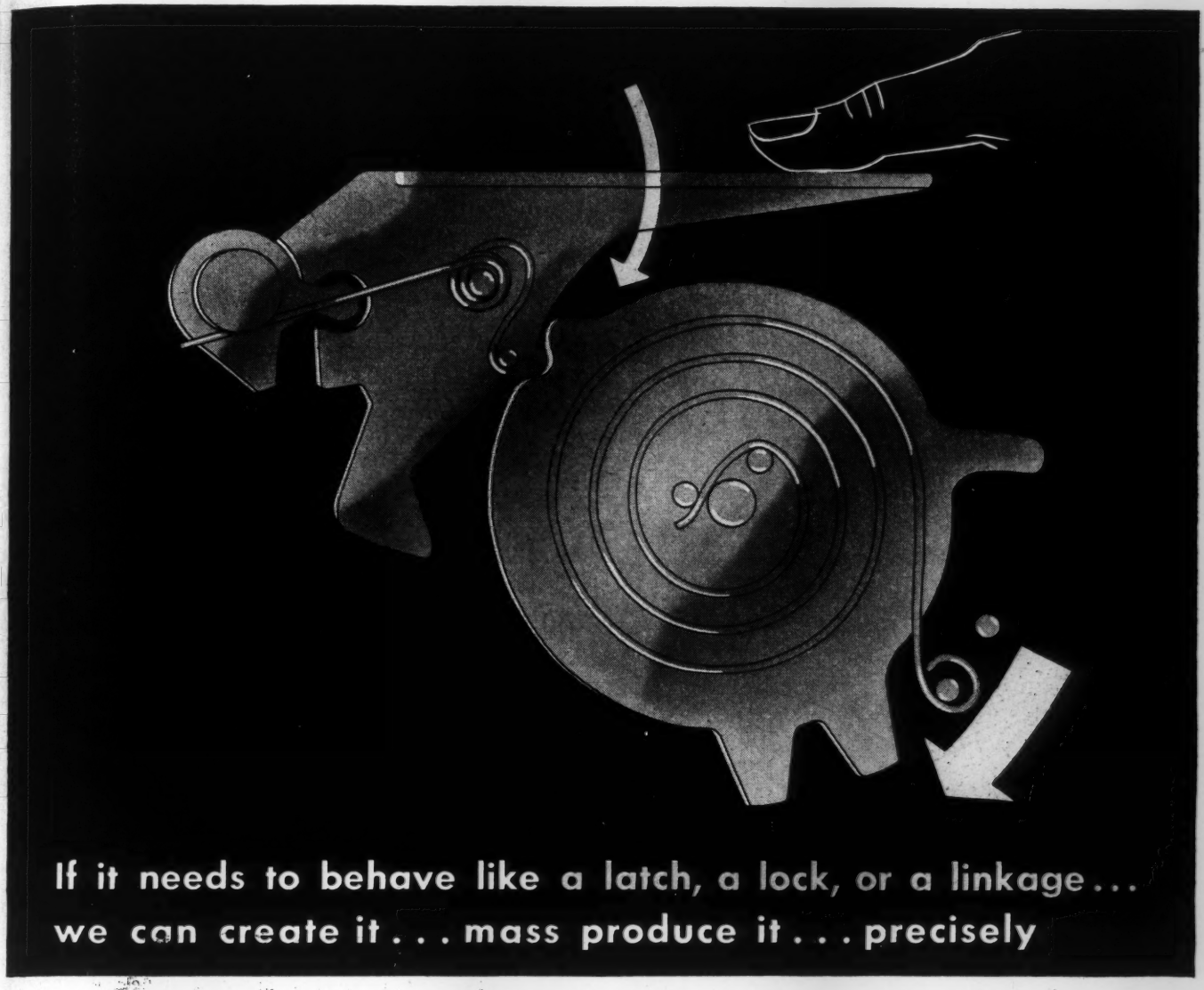
According to a recent announcement, **Chester F. Conner,** manager of industrial products sales of the B. F. Goodrich Co. and a widely known figure in the rubber industry, has been placed on the company's retirement payroll and several new appointments made in the organization. **Paul W. Van Orden** has been named industrial products merchandise manager; **Richard G. Cox** is now manager of original equipment sales; **Wilfred A. Smith** has been made manager of molded, extruded, lathe cut and sponge sales; and **George J. Fischer** was appointed manager of V-belts and packing sales.

♦

Electrol Inc., Kingston, N. Y., has announced the appointment of **Vincent A. D'Aprile** as sales engineer to handle the company's line of hydraulic devices in the industrial field.

♦

John H. Burch was recently appointed sales manager of the Detroit district of the Wickwire Spencer Steel division of the Colorado Fuel and Iron Corp. He will make his headquarters at the Detroit district sales office, 1915 National Bank Bldg., Detroit 26, Mich. Mr. Burch succeeds **L. A. Watts,** who will now devote his



If it needs to behave like a latch, a lock, or a linkage... we can create it... mass produce it... precisely

...WITH STAMPINGS!

If a part of your product needs to snap, latch, catch, link, lock, spring or trip—if it needs a mechanical motion to promote *safety*, convenience, performance or salability, Standard is the place to go.

We can make these devices so that they will work by spring, cam or lever. We can make them rigid or freely operated . . . to slide, swing, rotate or to produce any one of an endless variety of motions.

Sometimes they must be concealed, or withstand excessive vibration, or be tamperproof. For example, we have built a great deal of extra *safety* into the door locking devices of many, well-known automobiles.

Out of our experience with light and heavy stampings we have developed a variety of techniques—more modern stamping methods—more closely controlled assembly methods—to hold such devices to unusually close tolerances.

Does this careful approach to latches, locks and linkages suggest a way to improve your product? If so, we'd be pleased to discuss your application . . . any time.

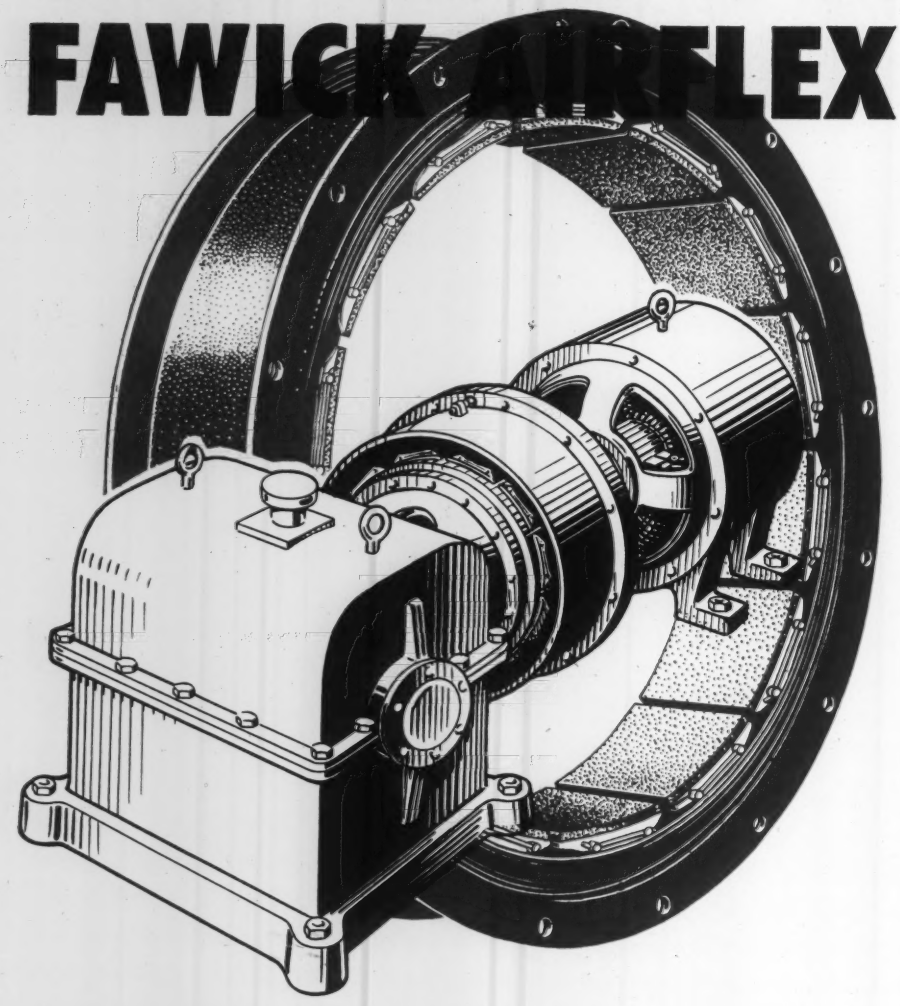
THE REID DIVISION OF *The Standard Products Co.*

THE MARK OF A SUPERIOR PRODUCT

DEPT. C, GENERAL OFFICES: 2130 WEST 110 STREET • CLEVELAND 2, OHIO

WE MAKE MOTIONS

Clutch with a DOUBLE PURPOSE FAWICK AIRFLEX



Fawick Airflex Clutch units serve a double purpose when installed as couplings between driving and driven machine members. They function equally well as disconnect couplings—where dependable engagement and disengagement are necessary, or as flexible couplings to absorb load shocks and dampen shaft vibrations.

In disconnect coupling service the fast, full-power action of Fawick Clutches has increased production on thousands of installations. Mechanical efficiency is also increased by the elimination of downtime for clutch maintenance, adjustment or lubrication.

These are a few reasons why you get top performance from your machines when you have Fawick Clutches serving the dual purpose of flexible couplings and disconnect couplings.

For specific information on all advantages of Fawick Clutch and Brake units, write to the Main Office, Cleveland, Ohio, for Bulletin 300.

All desirable clutch characteristics are built into Fawick Airflex units

FAWICK Airflex CO., INC.
9919 CLINTON ROAD CLEVELAND 11, OHIO

FAWICK CLUTCHES = AXIAL CONTACT + 360° RADIAL CONTACT + SHOCK ABSORPTION + CONTROLLED TORQUE = PEAK EFFICIENCY

full efforts in the capacity of sales manager of the wire and pig iron and semifinished steel departments of the Wickwire Spencer Steel division. Mr. Watts will continue to be located at the division's offices in the Wickwire Bldg., 361 Delaware Ave., Buffalo 2, N. Y.

♦

Formerly a service engineer for Kennametal Inc., Latrobe, Pa., **Floyd Monteith** has been appointed as representative working out of the company's Chicago office. Also named representative, **Robert Karakoosh** will work out of the Springfield office, as will **William Collins** and **Harold D. Killmer,** who have been appointed service engineers. **Conrad Seim** has been named service engineer to work out of the company's Los Angeles office.

♦

With headquarters at Plainfield, **Lawrence H. Russell** has been appointed to the position of sales manager of the Walker-Turner division, Kearney & Trecker Corp., Plainfield, N. J. Mr. Russell comes to his present position with ten years of experience in the power tool field, three years as sales manager of the Power King division of the Atlas Press Co.

♦

Lester Castings Inc., 3170 West 33rd St., Cleveland, O., has announced the appointment of **Ernest F. Karbe** as sales representative and service engineer for the state of New York. This appointment is part of an expanding program of field engineering service by Lester Castings for users of zinc and aluminum die castings.

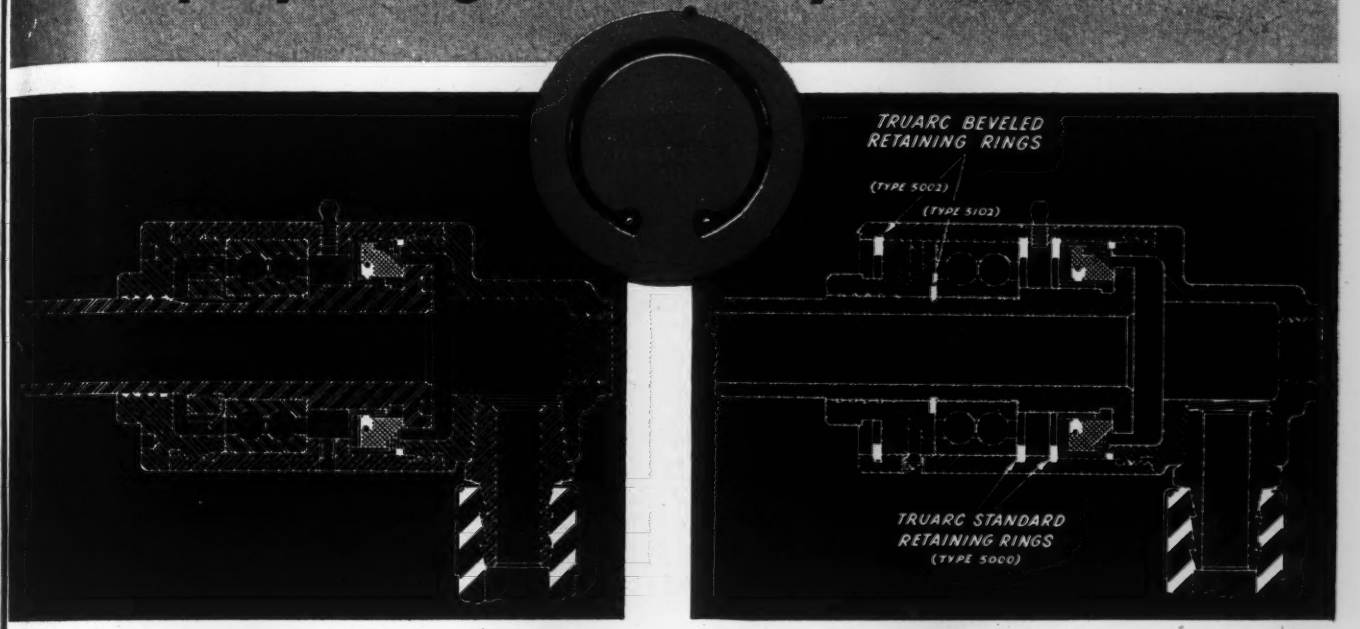
♦

Two new appointments were recently announced by La Salle Steel Co., Hammond, Ind., manufacturers of cold-finished steel bars. **M. G. Sladek** has been promoted to district sales supervisor and will take charge of accounts in Minnesota, Iowa, Wisconsin and most of Illinois and Ohio, supervising the activities of six men in the territory. **Paul L. Bisdorf,** a recent addition to La Salle's engineering staff, will be available for consultation on machining problems.

♦

Complete reorganization of its sales force, under the direction of **Orrin G. Meyers,** sales manager, has been announced by the Hunter Spring Co. of Lansdale, Pa. The new appointments and allocation of territories are as follows: **Bill Sohl** will handle western Pennsylvania and eastern Ohio; **Del Bennett,** New York State except New York City; **Sam Whiteley,** eastern Pennsylvania (except Philadel-

4 Truarc Rings save $2.41 unit cost, Simplify Design, Assembly, Maintenance



Conventional Way. This design died on the drawing boards, because the Deublin Company, Northbrook, Ill., found the simplified design, using Truarc Rings, superior and more economical.

Truarc Way. Truarc Beveled Retaining Ring holds cap in place, takes up end play, locks entire assembly. Second Beveled Ring positions ball bearing on rotor. 2 standard Truarc Rings position bearing and removable seal, eliminate shoulders, permit manufacture of housing from strong, non-porous, easily machined brass tubing. Unit can be serviced in minutes simply by removing 2 Truarc Rings.

HOW $2.41 WAS SAVED

Materials saved	
decreased wall thickness of housing	$.79
eliminated bearing lock nut and washer	.18
	.97
Machine operations eliminated	
Bore, undercut, and tap cap end of housing	.39
Locate cap on arbor, and chase threads	.28
Drill spanner wrench holes	.10
Cut thread on rotor for lock nut	.09
Mill slot in thread for tang on lock washer	.18
Drill spanner wrench holes in rotor	.10
	1.14
Assembly operations eliminated	
Install lock washer, tighten lock nut, bend lug	.20
Assemble cap into housing	.10
	.30
TOTAL SAVINGS	$2.41

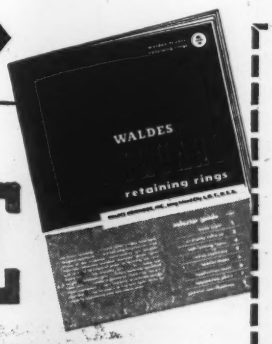
THE Deublin Union—a rotating joint for steam, air, or water—is simple, rugged, easy to service. 4 Waldes Truarc Rings hold entire unit together, permit simplified design, cut unit cost $2.41.

Improve and simplify your own product design with Truarc Rings, and you too will cut costs. Wherever you use machined shoulders, nuts, bolts, snap rings, cotter pins, there's a Waldes Truarc Ring that does a better job of holding parts together.

Truarc Rings are precision engineered. Quick and easy to assemble, disassemble. Always circular to give a never-failing grip. They can be used over and over again.

Find out what Truarc Rings can do for you. Send your blueprints to Waldes Truarc engineers for individual attention, without obligation.

SEND FOR NEW CATALOG



WALDES

TRUARC

REG. U. S. PAT. OFF.

RETAINING RINGS

WALDES KOHINOOR, INC., LONG ISLAND CITY 1, NEW YORK

WALDES TRUARC RETAINING RINGS ARE PROTECTED BY THE FOLLOWING PATENT NUMBERS: U. S. PATENTS 2,382,948; 2,420,921; 2,411,761; 2,487,803; 2,487,802; 2,491,306 AND OTHER PATENTS PENDING

Waldes Kohinoor, Inc., 47-16 Austel Place MD112
Long Island City 1, N. Y.

Please send the new catalog on Waldes Truarc Retaining Rings.

Name_____

Title_____

Company_____

Business Address_____

City_____ Zone_____ State_____

4KW

Try this flexible, easily installed

"JOHN CRANE" CARTRIDGE-TYPE SEAL

Simple to install on production lines; easy to replace in the field.

End-face sealing; raised face washer means remarkably low friction.

Flexibility takes care of washer wear, shaft vibration and shaft end play.

Positively driven by notches in driving sleeve.

High production tooling on sizes for all shafts up to .650".

Millions now in use on small shafts.

The "JOHN CRANE" 6A Seal, specially developed for rugged low pressure applications such as centrifugal water pumps and rotary agitator shafts, is produced in tremendous quantities for the automotive and appliance fields. High production tooling for this carefully engineered seal makes unit costs low.

This 6A Seal has enabled many manufacturers to build their equipment at a lower cost. Users of oil closures and all kinds of rotary seals should investigate it to see if it can do the same for them.

The 6A seal is fully described in this new booklet. It will be a valuable addition to your engineering library. Write Dept. MD-11 1825 Cuyler Ave., Chicago 13, Ill.

JOHN CRANE

PACKINGS AND MECHANICAL SEALS
CRANE PACKING COMPANY
CHICAGO
Offices in All Principal Cities in United States and Canada

phia), Delaware and Maryland; **James P. Glidden,** Massachusetts, New Hampshire, Vermont and Maine; **Donald A. Beardsworth,** Connecticut, Rhode Island and New York City; **Joseph E. Curry,** New Jersey; and **John Sigafoos,** Philadelphia County.

♦

Russell, Burdsall & Ward Bolt and Nut Co., Port Chester, N. Y., has appointed **Emmett F. Harding** as Port Chester sales manager, in addition to his present duties as manager of screw sales, with **Henry McCarthy** as assistant Port Chester sales manager. These changes resulted from the recent appointment of **Robert H. Burdsall,** former Port Chester sales manager, to the new post of manager of market development and advertising, in line with the company's program to co-ordinate promotional and sales activities.

♦

The appointment of **Irving R. Redland** as assistant to the chief engineer in charge of production and sales has beeen announced by Frazier & Son, 338 Cortland St., Belleville, N. J. Frazier & Son is the manufacturer of "Fra-sure" variable speed controls with automatic speed locking device for use with fractional horsepower motors and "Whiz Packer" package filling machines for dry, granular products.

♦

Lear Inc. of Grand Rapids, Mich., has announced the appointment of **C. E. Willis** as assistant sales manager of its Electro-Mechanical division and **A. N. Lawrence** as manager of its Teterboro, N. J. Eastern office. Mr. Willis has been manager of Lear's Eastern office since 1947. Upon assuming his new duties of directing West Coast sales and field engineering activities, he will make his headquarters at the company's Los Angeles plant. Mr. Lawrence, formerly aviation application engineering department manager for Jack & Heintz, brings to Lear a successful background in aircraft equipment sales engineering.

♦

Three major changes in executive personnel were announced recently by the Laminated Shim Co. Inc. **Richard Seipt,** who has been with the company since 1935 and has served as vice president in charge of sales for ten years, will function in the future as vice president and general manager. **Merle L. Lockwood** has been appointed vice president in charge of sales to succeed Mr. Seipt, and **Michael E. Errico,** formerly assistant sales manager, has been appointed sales manager.

All these Valves ... from One!

Valve combinations for 90% of industrial piping assembled with 4 bodies and a handful of parts.

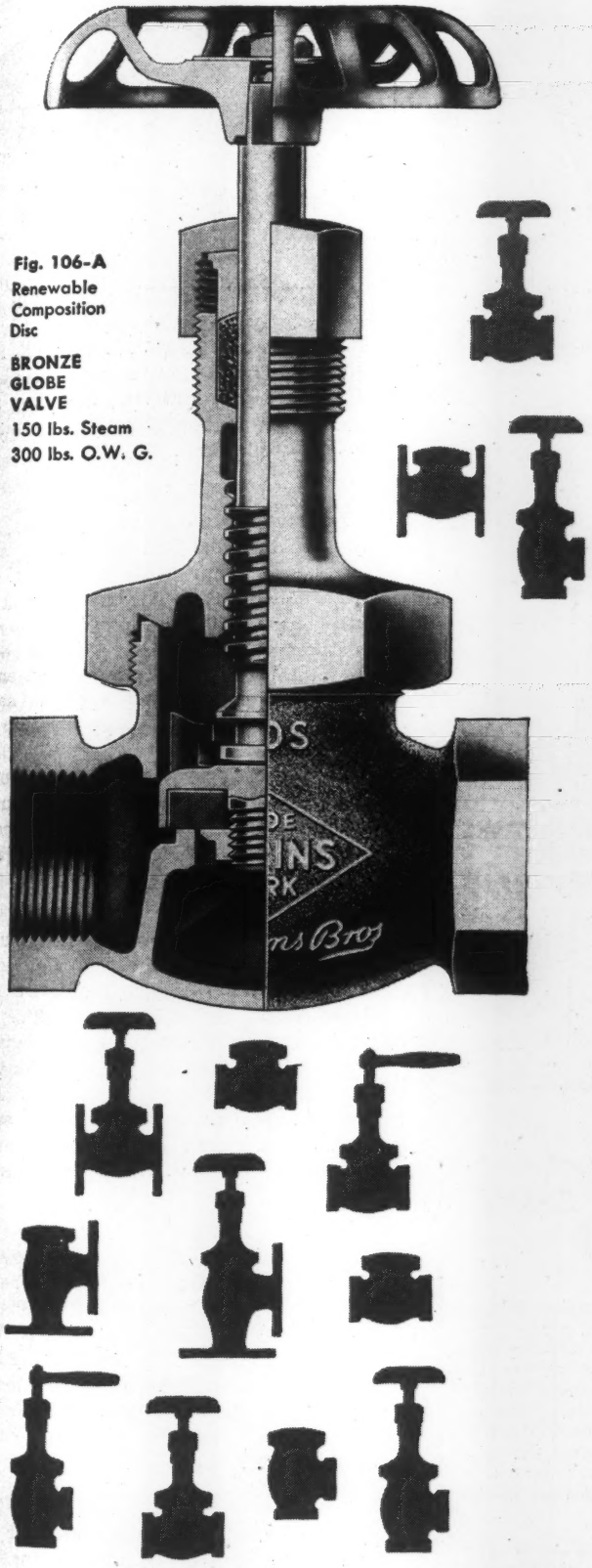
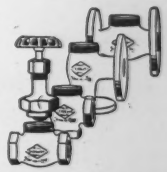


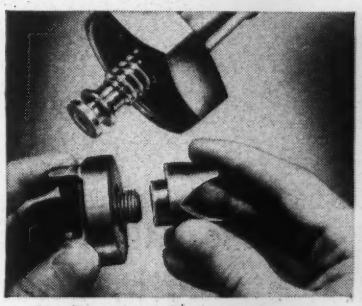
Fig. 106-A
Renewable Composition Disc

BRONZE GLOBE VALVE
150 lbs. Steam
300 lbs. O.W. G.

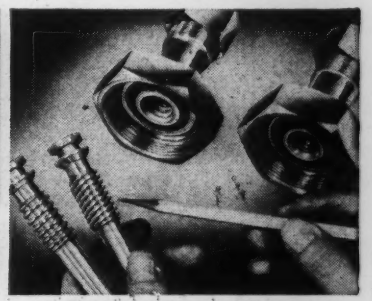
JENKINS FIG. 106-A "FAMILY"

Start with the standard Fig. 106-A. Trimming is interchangeable in Globe or Angle body, screwed or flanged.

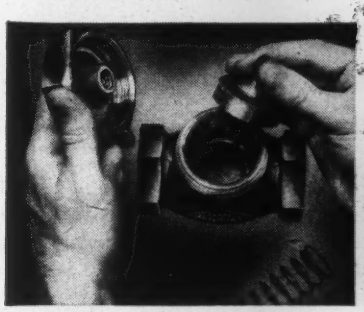




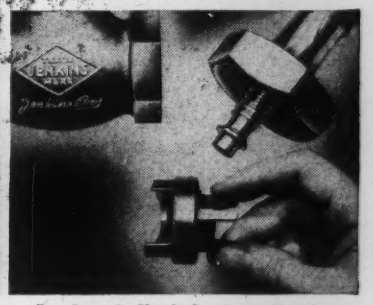
For Close Control — In throttling service, the nut which holds the disc in the disc holder is removed and replaced with Throttling Nut, Fig. 344.



For Quick Opening and Closing — Merely substitute the bonnet and spindle from Fig. 941, in which threads are pitched more sharply.



For Lift Check Service — Globe or Angle Bodies can be fitted with interchangeable Cap, Disc Holder, and Guide Disc Nut from Fig. 117A. Addition of spring from Fig. 655A provides spring loaded service.



For Stop & Check Service — Use the 106A trim, but substitute this spindle from Fig. 630A and replace the regular disc nut with the check valve guide disc nut.

Look over these pictures and see how easy it is to make up just the valve type you need, simply by interchanging parts. Think of the saving this means in maintenance expense . . . in reduced inventory of spare valves in your stockroom.

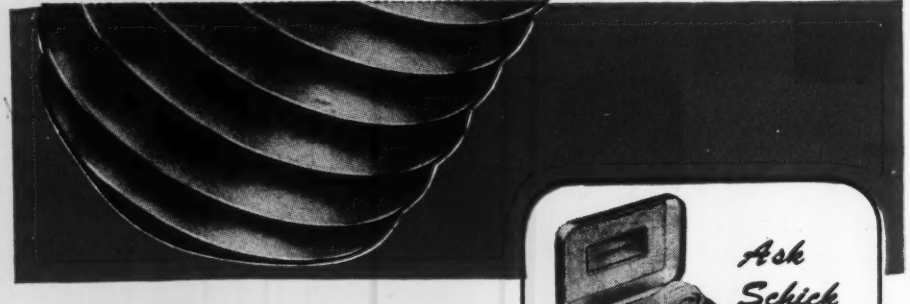
That's a big reason why the Fig. 106-A series takes top honors in any poll for valve preference. And Jenkins time-proved design, makes them the longest-lasting, lowest-upkeep valves that money can buy.

For example, see the heavy construction of the one-piece, screw-over bonnet. You can remove and replace it over and over again without distortion. See the extra size packing box,—and the perfectly machined threads on the heavy manganese bronze spindle, with more threads in contact with the bonnet, open or closed.

Remember, Jenkins Bros. is the originator of the renewable composition disc valve,—the *only* manufacturer of both valves *and* discs. For drop-tight, trouble-free performance, get acquainted with the Fig. 106-A "family". Despite their extra value, *you pay no more* for Jenkins Valves. Jenkins Bros., 100 Park Ave., New York 17, N. Y. Jenkins Bros., Ltd., Montreal. Sold through leading Industrial Distributors.

JENKINS VALVES

In ANY size get genuine ALLEN ⬡ HEAD screws
. . . . for greater strength, accuracy, grip



If you buy for production or replacement . . . Be sure to get genuine Allen ⬡ Head screws in the familiar black and silver striped box. Used with genuine Allen keys and drivers they give you the unbeatable fastening team.

To help improve your product, we work constantly with manufacturers to create better products through more dependable fastenings. We invite you to write us if you have such a problem.

Ask Schick

Here's an application for a tiny #6 socket set screw developed by Allen to hold the laminated rotor to the vibrating shaft in a Schick Electric Razor. This Allen ⬡ Head screw, unlike the ordinary slotted type, insures the unfailing performance required by this leading shaver manufacturer.

SOLD ONLY THROUGH LEADING DISTRIBUTORS
Write the factory direct for technical information and descriptive literature.

WARNING
Allen-Type screws aren't necessarily Allen-Made.
Get genuine "Pressur-Formed" Allen ⬡ Head screws in this black and silver box.

ALLEN ALLEN HEAD
MANUFACTURING COMPANY
Hartford 2, Connecticut, U. S. A.
NEW YORK, CLEVELAND, DETROIT, CHICAGO, LOS ANGELES

SALES NOTES

ANNOUNCEMENT was made recently of the appointment of **Bates Sales Co.**, 1025 North Sixth St., St. Louis 1, Mo., as factory sales and service representative in the St. Louis area for **Howell Electric Motors Co.**, Howell, Mich.

♦

Raybestos-Manhattan Inc. has removed its New Orleans office and warehouse from 1009 Camp St. to 920 Calliope St. The new facilities will enable the company to serve customers in that area with a more complete stock of belting, V-belts, hose, other industrial rubber products, packings and asbestos textiles.

♦

The Detroit office of the **Denton & Anderson Co.** has moved to a new location at 520 West Eight Mile Rd., Ferndale, Mich.

♦

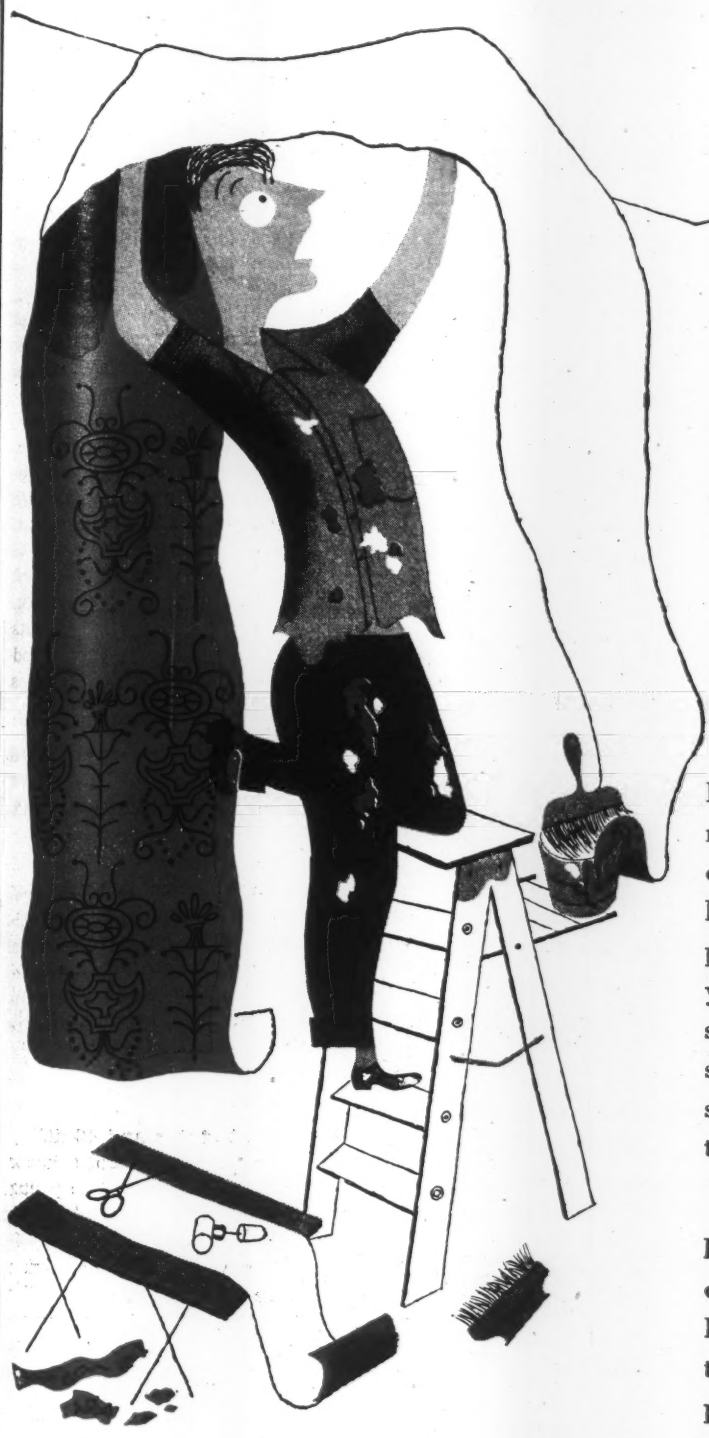
To serve growing Midwestern business, **George H. Elliott & Co.**, consulting management engineers of New York, has opened an office at 606 Bankers Trust Bldg., Indianapolis. The new office will be under the direction of Robert T. Ramsey Jr., who headed his own firm of consulting management engineers prior to his affiliation with the Elliott organization.

♦

A new sales office for the **General Electric Company's** chemical department has been opened in the Shoreham Bldg., 806 Fifteenth St., N. W., in Washington, D. C. Products of the department include silicones, laminated and molded plastics, electrical insulation, plastics molding compounds, and resins.

♦

Long identified as a producer of hydraulic control equipment for industry and aviation, **Adel Precision Products Co.** has changed its corporate identity to **Adel Division, General Metals Corp.** The change became effective upon the consolidation of Adel with two other nationally known manufacturing concerns, **Enterprise Engine and Foundry Co.** of San Francisco, and **General Metals Corp.**, with California plants located at Los Angeles and Oakland and a third at



some things need experience

Putting stainless steel to work in a given application requires experience. Since stainless is a whole family of steels, the *right* analysis must be used . . . or stainless steel will not function properly. That's why Crucible, pioneer in the development of this specialty, offers you, freely, the experience of an alert metallurgical staff. These engineers and metallurgists know which stainless fits your job best . . . and their advice can save you time and money in applying stainless steel to your requirements.

The painstaking care Crucible shows in finding the best answer to Industry-posed problems is the measure of Crucible's half century of specialty steel leadership. Put this experience to work for you. And, whether the order is in tons or pounds, your problem is assured prompt attention. When it's stainless...call in Crucible. CRUCIBLE STEEL COMPANY OF AMERICA, Chrysler Building, New York 17, N. Y.

CRUCIBLE first name in special purpose steels

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fifty years of Fine *steelmaking*

STAINLESS • HIGH SPEED • TOOL • ALLOY • MACHINERY • SPECIAL PURPOSE STEELS

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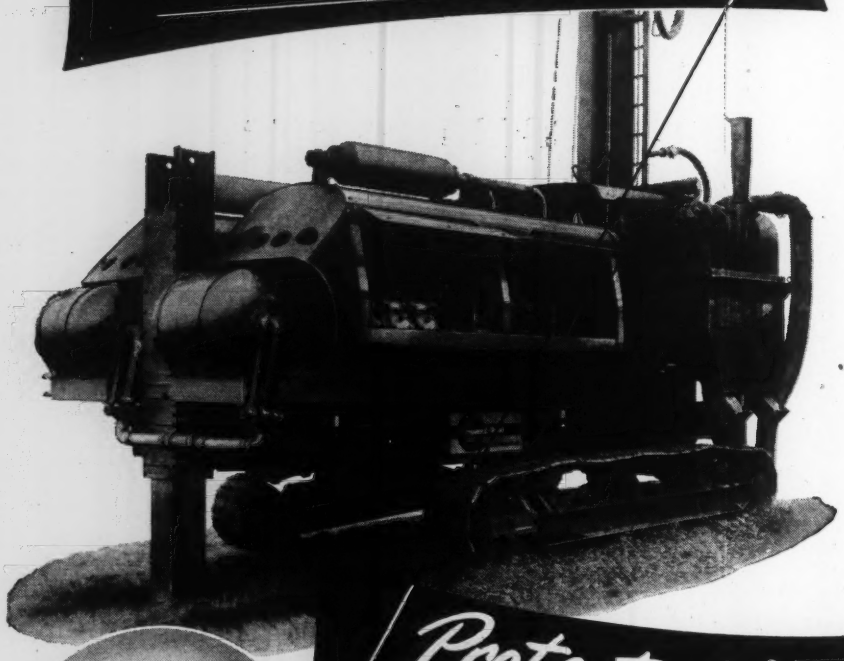
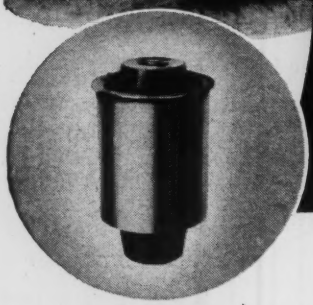




Protected by LORD MOUNTINGS

Compressors must give dependable service under gruelling conditions of rough terrain, mud, and constant exposure to the shock and vibration of drilling. Unless protected from shock and vibration, even the most rugged equipment is subject to excessive down-time and maintenance.

Ingersoll-Rand Company solves this problem in its rugged mobile "Quarrymaster." Each of the 10,000-pound compressor power plants is protected by four LORD Bonded-Rubber Mountings which prevent shock and vibration from interfering with efficient performance. They also prevent excessive twisting stresses in the power plant when the "Quarrymaster" is moving over uneven ground.

Whether your product is large or small, reduction of shock and vibration improves performance and accuracy . . . lowers maintenance costs . . . increases customer satisfaction. Investigate LORD Vibration Control Mountings and Bonded-Rubber Parts. Write to attention of Product and Sales Engineering Department.

LORD MANUFACTURING COMPANY, ERIE, PA.

Canadian Representative: Railway & Power Engineering Corporation, Ltd.

LORD BONDED RUBBER — *Vibration-Control Mountings . . . Bonded-Rubber Parts*

Houston, Tex. It was also announced that Fred T. Miller has been named vice president of the Adel division, and that the division has opened new offices in the Stevens Bldg., 42 Third Ave., Mineola, Long Island, N. Y., closing the offices at Stamford, Conn. J. Perry Barker will be in charge of the Mineola office, continuing to act as Adel's eastern representative, and E. M. Storer Jr. will join the organization as a sales engineer.

♦

To handle Barry equipment for shock and vibration control in the Southwestern territory, R. M. Campion, **Campion Sales Co.**, has been appointed representative by the **Barry Corp.**, 175 Sidney St., Cambridge, Mass. The Campion company is located in the Republic National Life Bldg., P. O. Box 4116—Station A, Dallas 8, Tex.

♦

The **Durant Manufacturing Co.** of Milwaukee has recently opened a southern branch office on Route 7, Greenville, S. C. Harold V. Barker has been appointed manager of the new branch.

♦

A branch office at 214 West Tenth St., Wilmington 1, Del., has been opened by the **Foxboro Co.**, Foxboro, Mass. This new office is under the managership of Kenneth L. Barton, who has been with the company for twenty-five years. Earlier this year a Foxboro Co. office was opened at 225 South Fifth St., Minneapolis 2, Minn., with Robert C. Cahill as resident engineer.

♦

In 23 different states and 40 different engineering schools, both honor and cash have been awarded to young engineers who were successful in the 1949-50 **James F. Lincoln Arc Welding Foundation's** annual engineering undergraduate award and scholarship program. A total of $5000 was awarded in 77 cash awards, and $1750 was given in scholarship funds to schools.

♦

The **Weatherhead Co.**, Cleveland, has announced the formation of its new industrial sales division, with **A. D. Hodge** as sales manager. A carefully planned sales and service program has been formulated for the new division, handling the complete Weatherhead industrial line—high-quality hose and reusable steel hose ends; Ermeto fittings in steel, stainless steel and brass; and standard SAE flare, compression, inverted flare, threaded sleeve and pipe fittings (available in brass). Under the new program Weatherhead industrial

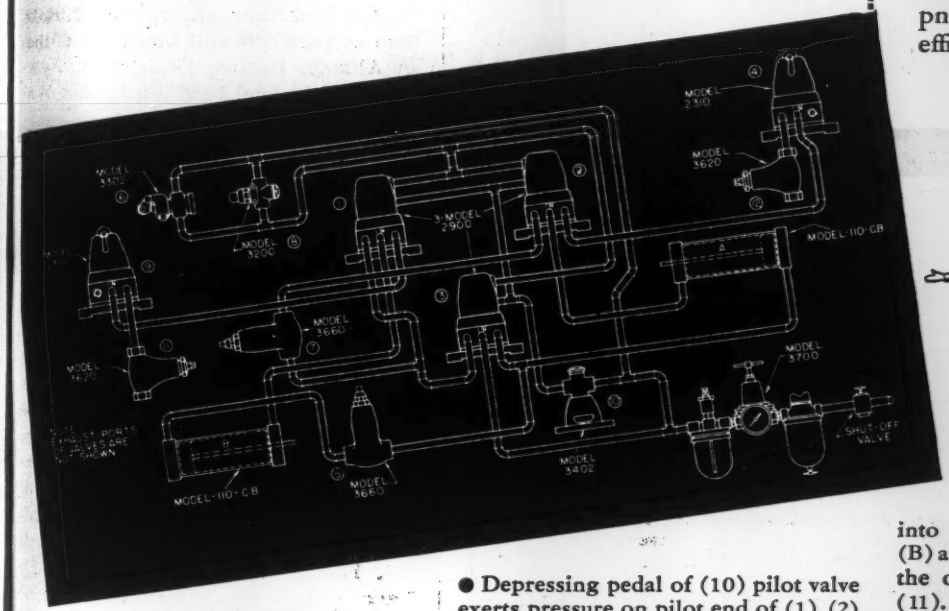
WHAT MAKES UP A Good Air Circuit?

1 VALVES, CYLINDERS OF SUPERIOR DESIGN

Rivett meets this requirement with equipment designed by foremost engineers in the air field. Compactness of 4-way valves prevents fouling or leaking; large, convenient pipe connections allow unrestricted air flow. Cylinders are notable for absence of tie rods. Keeper ring design permits compact installation.

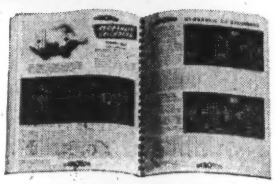
2 KNOWLEDGE OF APPLICATION

Equally important is the know-how as demonstrated by Rivett Factory and Distributor Representatives. These men possess the knowledge and experience to help you draw up the most suitable pneumatic circuit to furnish efficient and lasting service.



AIR EQUIPMENT ON AN AUTOMATIC LATHE FOR CLAMP, TURNING AND CUT-OFF

● Depressing pedal of (10) pilot valve exerts pressure on pilot end of (1), (2) and (3) pilot operated 4-way valves. Air flows from valve (3) to blind end of clamp cylinder (A). Piston moves out rapidly, the air from rod end being exhausted thru (4) cam operated 4-way valve. When clamp cylinder rod (A) depresses cam of valve (4), exhaust air from cylinder is restricted (12), providing for a slow approach. When clamp cylinder engages work, pressure in supply line increases, causing sequence valve (5) to open. Air flows into blind end of cut-off tool cylinder (B) and a similar action takes place thru the out-stroke, by means of (9) and (11). After the cut-off, cylinder rod depresses cam of (6) pilot valve which, by pilot pressure, reverses valves (1), (2), and (3). Air line is then connected to the rod end of cylinder (B) and retracts cut-off tool. After cylinder completes return stroke, pressure builds up in supply line connected to sequence valve (7). Pressure opens valve (7) and allows air to rod end of clamp cylinder. After it retracts, cycle is completed. Pilot valve (8) permits interruption of cycle to retract cylinders.



Get Rivett Catalog Sections No. 54 and No. 302
describing air valves and cylinders

Write to
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WHEN YOU APPLY HYDRAULIC OR AIR POWER *Plan with*

RIVETT

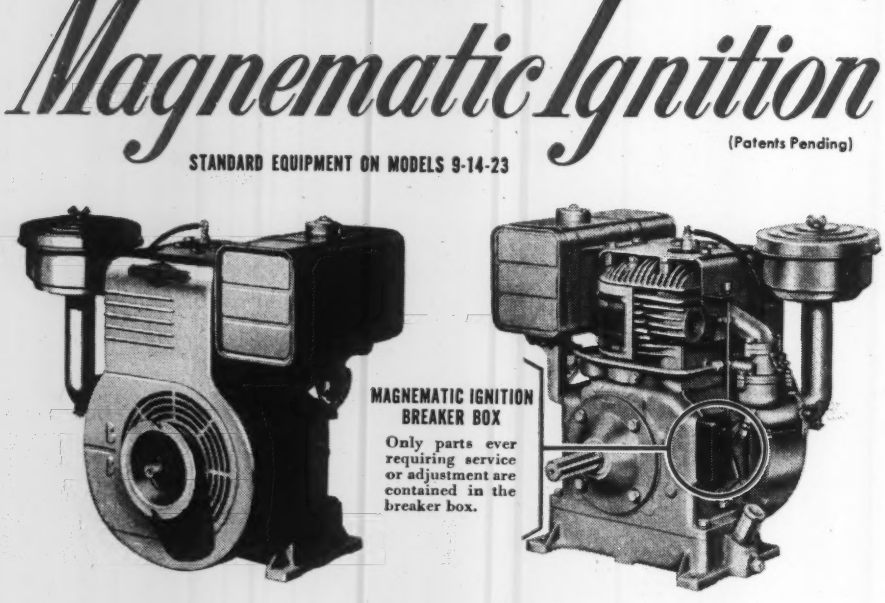
AIR AND HYDRAULIC VALVES AND CYLINDERS, HYDRAULIC POWER UNITS

ONLY BRIGGS & STRATTON
HAS THE MODERN

Magnematic Ignition

(Patents Pending)

STANDARD EQUIPMENT ON MODELS 9-14-23



A Dependable Spark For Better Engine Performance at ALL SPEEDS

Magnematic Ignition delivers a dependable spark at lower speeds and better engine performance at all speeds.

LONG SPARK PLUG LIFE — magneto output tailored to engine requirements — sparks only on firing stroke — voltage decreases as speed increases, substantially lengthening spark plug life.

AUTOMATIC SPARK ADVANCE — spark is retarded at starting speed and automatically advanced as engine speed increases.

QUICK, EASY STARTING — pull-up cranking with no danger of a "kick" during rope or hand cranking.

SEALED, WATERPROOF COIL — one piece, plastic insulated coil, fully sealed and waterproofed with molded-in high tension lead.

SIMPLE AND DEPENDABLE — no extra gears, bearings or lubrication system.

EASY TO SERVICE — only parts ever requiring service or adjustment accessible by removing breaker-box cover conveniently mounted outside crankcase.

Magnematic Ignition is another exclusive Briggs & Stratton development which is setting new standards of performance for single-cylinder, air-cooled engines.

Write for bulletin giving complete information on Magnematic Ignition and Briggs & Stratton engines.

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4 CYCLE BRIGGS & STRATTON GASOLINE ENGINES

Preferred Power FOR INDUSTRIAL CONSTRUCTION RAILROAD AND FARM EQUIPMENT

World Wide FACTORY SUPERVISED SERVICE

In the automotive field Briggs & Stratton is the recognized leader and world's largest producer of locks, keys and related equipment.

salesmen are strategically located in key cities to render the most efficient service possible and to establish closer contact with Weatherhead accounts from coast to coast. A special educational plan directed by Weatherhead factory trained sales and service engineers has also been included to acquaint distributor and jobber salesmen with all details of the Weatherhead line.

♦

The **Haloid Co.** of Rochester, N. Y., has announced the opening of a branch office at 814 Carnegie Hall, 1220 Huron Rd., in Cleveland. James R. Donovan, formerly of the Rochester headquarters, will be in charge of the new office.

♦

Tek Bearing Co. Inc., has announced the formation of a new corporation, **Tek Service Co. Inc.**, to more effectively serve customers' plant equipment and maintenance needs with a complete line of antifriction bearings and mounted units. The corporation will maintain offices in Albany, Boston, Bridgeport, Newark, Providence, New York, and Waterbury.

♦

The office of the **Resistance Welder Manufacturers Association** has been moved from 505 Arch St., Philadelphia, to newly remodeled and modernized quarters at 1900 Arch St. in Philadelphia.

♦

The **De Laval Steam Turbine Co.** of Trenton, N. J., has announced the appointment of the **R. M. Sorlie Co.** of Kalamazoo, Mich., as a sales representative and the appointment of **Robert F. Curry** to the De Laval office, Detroit. Both of the new offices will handle the full line of Imo-De Laval pumps, worm and herringbone gear speed reducers, and couplings.

♦

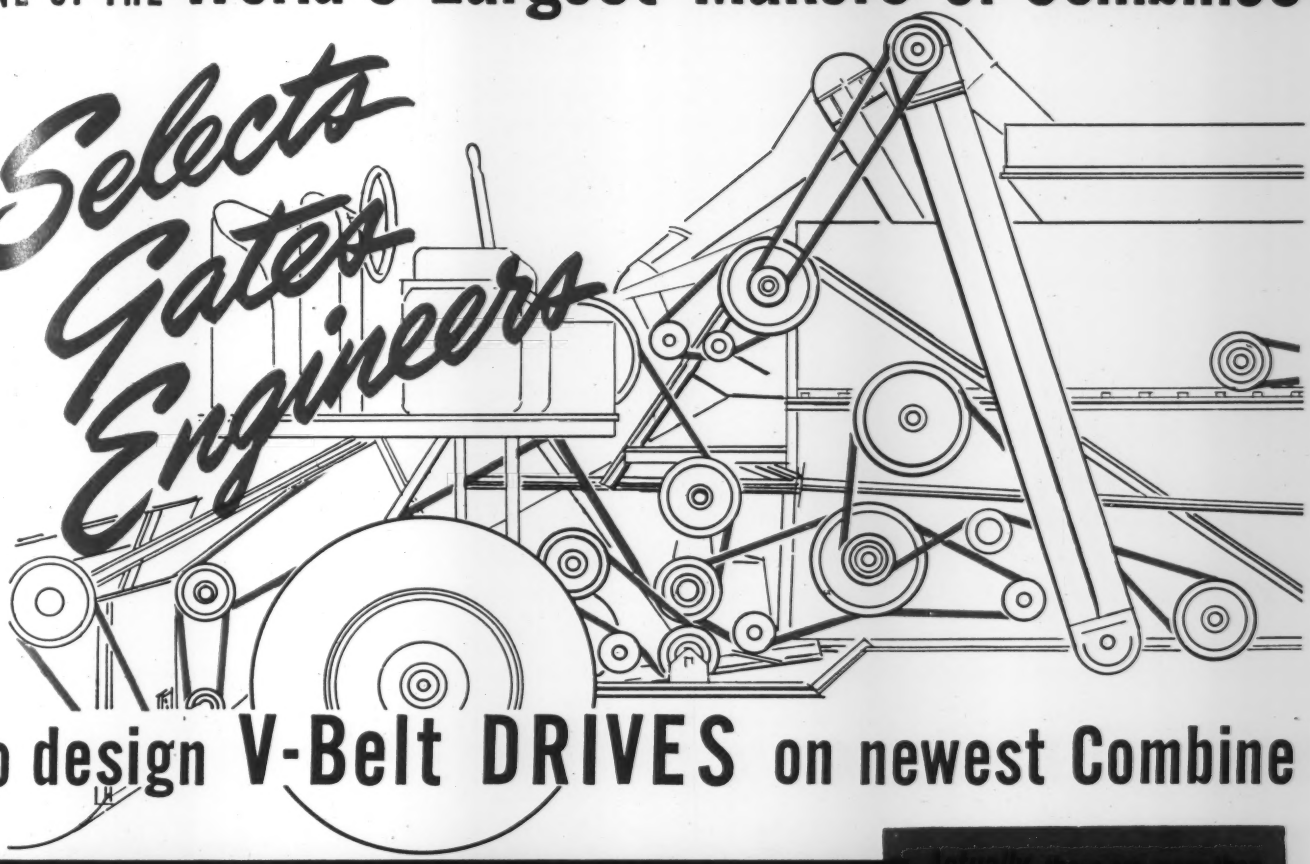
According to a recent announcement, the **Barium Steel Corp.**, 25 Broad St., New York, N. Y., has opened an office at 840 Connecticut Ave., N. W., Washington, D. C., to act as sales liaison with government agencies.

♦

The New York sales office of **B. F. Goodrich Chemical Co.** has moved into new and spacious facilities in the Farmers Loan and Trust Co. Bldg., 475 Fifth Ave. Occupancy of the entire twenty-fourth floor will provide additional offices and conference rooms necessitated by expanding operations.

ONE OF THE World's Largest Makers of Combines

Selects Gates Engineers



to design V-Belt DRIVES on newest Combine

A Completely V-Belt Driven Combine . . . Every drive of the most advanced design

In order to reduce the wear and tear of operating his combine, the manufacturer wanted every moving part that was subject to shock to be "cushioned" by the **shock-absorbing** properties of a V-belt Drive—and he wanted to be sure that every V-belt Drive embodied the most advanced engineering design.

He knew that the Gates V-Belt testing laboratories are by far the largest in the world—that Gates runs an average of 32,000 hours of testing every **week** on V-Belts alone! (32,000 hours is equivalent to five years of life for one belt.)

And he knew that Gates hadn't stopped with testing—that Gates Research Engineers had actually reduced the **results** of these tests to **usable data** for the design of every V-belt Drive.

This intensively SPECIALIZED Research has naturally given Gates Engineers a most advanced knowledge of V-Belts and V-Belt Drive Design—and the benefit of this advanced knowledge and experience is available to **you, right in your own plant, any time you want it!**

Whether you have a difficult drive to design—whether some V-Belt drive in your plant is giving trouble—or whether you merely want to know what size and construction of V-Belts will give the most efficient and the lowest cost service on a particular drive—you have only to phone a Gates Engineer.

Actually, there are more than a dozen GATES V-BELT DRIVES on this One Combine!

Some of them you can't see, no matter what view of the Combine is given. Every one of these drives had to be of the most advanced engineering design to insure superior service.

Gates Eng-509

The Mark of SPECIALIZED Research

There is always a Gates Field Engineer near you in all industrial centers. Just look in your phone book under "Gates Rubber." He will come to you promptly—and he will put at your service the full benefits of Gates unrivaled V-belt knowledge and experience **without the slightest obligation!**

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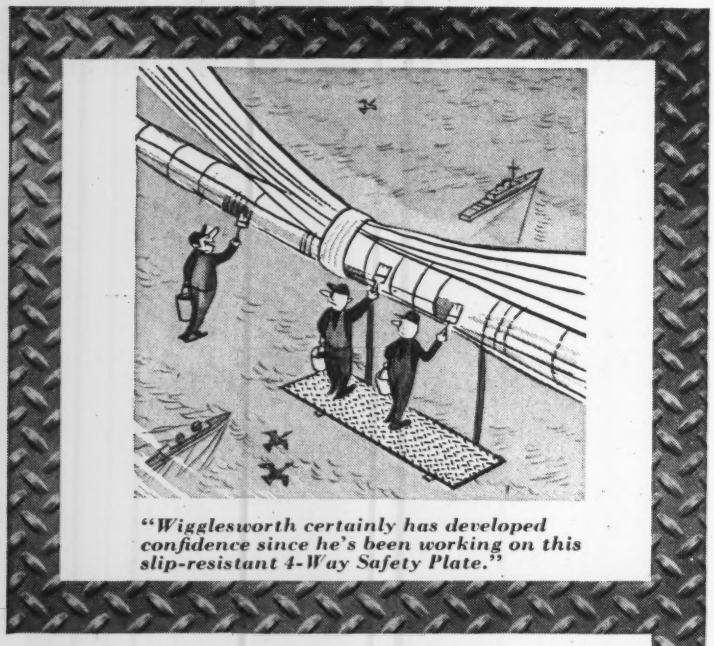
The World's Largest Makers of V-Belts
DENVER, U. S. A.

GATES VULCO ROPE DRIVES
Engineering Offices and Jobber Stocks IN ALL INDUSTRIAL CENTERS of the U. S. and 71 Foreign Countries



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For greater safety under foot,
in your plant and on your products

Inland 4-Way® Safety Plate

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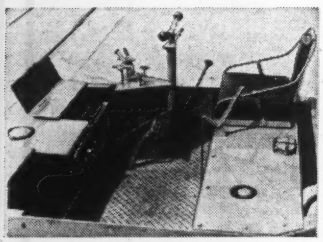
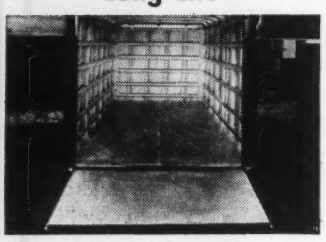


Long Life

Quick Starts—Stops



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INLAND

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NEW MACHINES

And the Companies Behind Them

Agricultural Equipment

TRACTOR: Small three-wheel tractor for utility farm or home use. Powered by 1½ or 2-hp Briggs and Stratton engine. Dimensions: width outside of rear tires, 30 in.; length overall, 48 in.; height, 26 in. Weight, 154 lb; speed, to 4½ mph. Small trailer truck and 2 or 3-gang mowers available as accessories. *Standard Mfg. & Sales Co., Lebanon, Ind.*

Domestic

GARBAGE DISPOSER: Redesigned units more economical to operate and 50 per cent quieter than previous models. Both units have new shredding mechanism with additional cutting action for improved efficiency. Fit most sink installations having 3½ to 4-in. drain outlets. *General Electric Co., Bridgeport, Conn.*

Earthmoving

EXCAVATOR: Combination digger and loader attachment for Ford and Ferguson tractors. Tractor-powered, hydraulically operated unit digs and shovels to 7-ft depth, back hoes by reversing shovel, digs plumb trenches on side hills to 15-degree grades, back hoes and front hoes 22-in. wide trench, elevates bucket 10 ft above grade and swings load laterally in 110-degree arc. Operated by four lever controls on tractor. Capacity of shovel bucket, 5 cu ft. *Pippin Construction Equipment Inc., White River Junction, Vt.*

Heating and Ventilating

PROJECTION UNIT HEATERS: Redesigned and enlarged line with capacities from 55,300 to 550,000 Btu per hr. Only motor and housing brackets protrude above top of unit. Louver Cone diffuser adjustable to direct air in any desired pattern from ceiling mounted units. *The Trane Co., La Crosse, Wis.*

APARTMENT HEATERS: Small gas heating plants designed for individual apartment use. Heats 3 to 5 rooms. Input rating, 45,000 Btu; weight, 160 lb; dimensions, 13 by 26 by 17 in. Unit can be suspended from ceiling or placed on shelf. Fully automatic unit requires no electrical connections. *Hook and Ackerman Inc., New York, N. Y.*

Heat Treating

GEAR HARDENER: Radio frequency machine for high production heat treating of gears. Used with vacuum-

twenty typical GRAMIX® parts that save time, money and machining

—die pressed of powdered metal mixtures . . . oil impregnated for self-lubrication

nickel silver pump rotor 30c
bronze saddle bearing 18c
brass fare token 1c
record player 2c center post
iron pulley 7c
contact tip 1½c
plate spacer 3½c
iron hinge bracket 17c
bronze clock bearing ½c
silver graphite contact 23c
iron sprocket gear 3c
heat-treated iron key 2½c
pure copper contact 3½c
bronze roller bearing retainer 4c
heat-treated drive gear 9½c
iron cam 5½c
dairy metal 23c pump liner blank
iron pump rotor 7c
iron clutch segment 2½c
iron ratchet pawl 9½c

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tube radio frequency generator and 10,000-cycle motor generator set. Inductall machine carries each gear through automatic cycle for through or contour hardening. Automatic loading and unloading magazines available. *Westinghouse Electric Corp., Pittsburgh, Pa.*

Manufacturing

AUTOMOTIVE PRESSES: Line of four-point, underdriven eccentric presses for automotive stamping plants. Sixty-three standard sizes with capacities from 300 to 3000 tons, bed areas from 72 by 48 in. to 300 by 108 in. Bearings arranged to permit shafts to be dropped straight down. Presses are double geared with driving eccentrics cast integral with main drive gears. *E. W. Bliss Co., Detroit, Mich.*

CORE BAKING TURNTABLE: Single unit with all components in one housing. Electronic unit generates heat within cores, keeps core room cool. Capacity, 350 lb of sand cores per hr. Handles cores to 15 in. wide, 28 in. long, with max combined height of core and core plate of 8 in. Turntable speed adjusts between 0 and 6 fpm. Requires 220, single-phase, 60-cycle current. Unit rated at 12 kva. *Induction Heating Corp., Brooklyn, N. Y.*

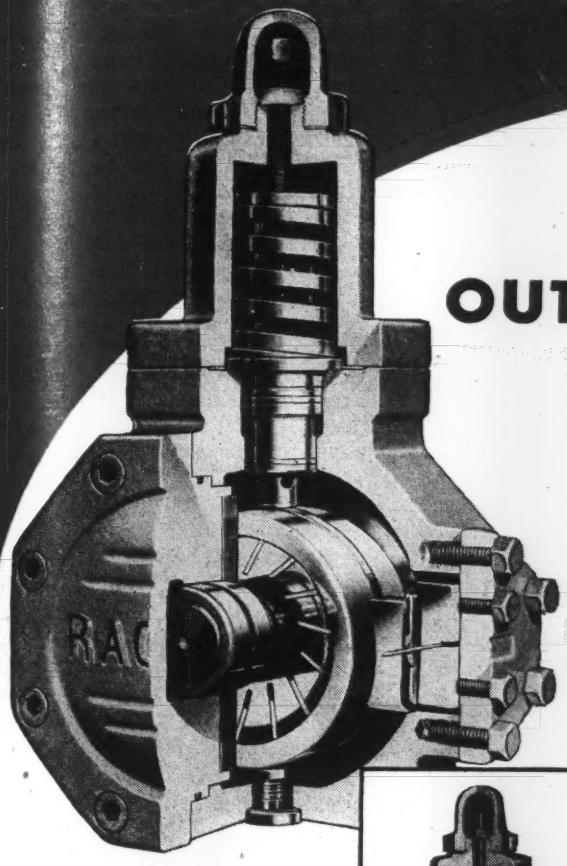
DRILL RIVETER: Drills and countersinks hole, feeds and heads rivet. After work is positioned, machine automatically performs all four operations when foot switch is depressed. Work clamped during work cycle, producing accurate holes without burrs between work pieces. Standard model uses max rivet length of 1⅛ in. Max force of 16,000 lb heads 5/16-in. solid aluminum alloy rivet. Production rate, 15 fastenings per minute. *General Riveters Inc., Buffalo, N. Y.*

MILLING ATTACHMENT: Quill travel vertical-mill attachment for bench, light floor or heavy mills. Allows full use of table travel, features quick, rigid angular settings. Spindle speeds of 380, 700, 1200, 2500 and 5200 rpm. Quill travel, 1½ in. Includes locking micrometer depth stop graduated to 0.001-in. *Rusnok Tool Works, Chicago, Ill.*

WIRE STRIPPER: Uses frictionally generated heat to melt enamel and Formex type insulations from ends of wires. New model with vacuum base handles wire sizes 48 through 25. Stripping wheels can be redressed as they wear. Weight, 38 lb; table space required, 10 by 15 in. *Rush Wire Stripper Div., The Eraser Co. Inc., Syracuse, N. Y.*

JIG BORER: Type 5 machine has max boring capacity of 9 in., vertical travel of boring spindle of 9 in. Accuracy of slide location guaran-

Put ALL of the Oil to Work

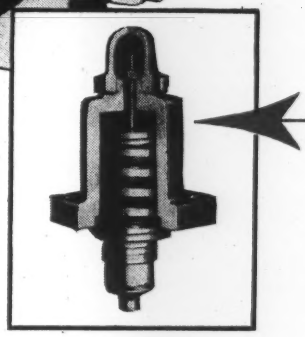


Get the Most

OUT OF YOUR CIRCUITS

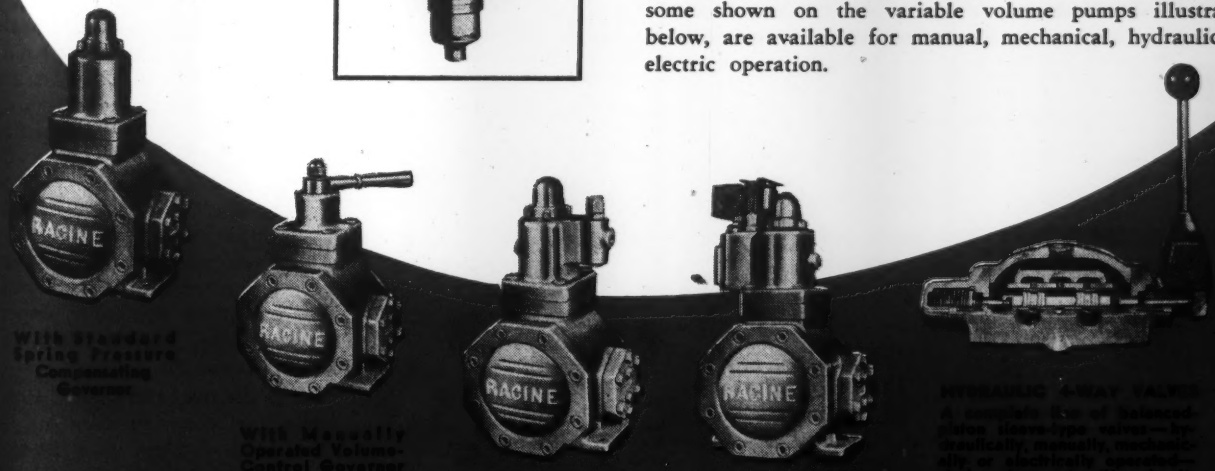
Top circuit efficiency demands effective regulation of volume and pressure. RACINE Pumps are all equipped with built-in governors — an exclusive RACINE feature. These governors regulate pressure and control the oil flow, delivering *only* the volume needed for the work of the circuit. They "put all of the oil to work." No by-passing — no relief valves — heating is reduced — horsepower conserved.

Governor-controlled RACINE Pumps give you efficiency, flexibility, and economy over a wide range of circuit conditions. Why not review the advantages "Variable Volume" offers for machines in your plant and for your product.



Interchangeable Governor Controls

At left is illustrated a standard pressure-compensating governor used in RACINE Pumps. Many other governors, some shown on the variable volume pumps illustrated below, are available for manual, mechanical, hydraulic or electric operation.



With Standard Spring Pressure Compensating Governor

With Mechanically Operated Volume-Control Governor

With Hydraulically Operated 2-Pressure Governor

With Solenoid Operated 2-Pressure Governor

HYDRAULIC 4-WAY VALVES — A complete line of balanced-piston sleeve-type valves—hydraulically, manually, mechanically or electrically operated—designed to simplify circuits and to meet an extensive range of operating conditions. Available in sizes ⅜" to 1½" IPS.

Write for your copy of our new 3-color Hydraulic Equipment Catalog. Address Racine Tool & Machine Co., 1773 State Street, Racine, Wisconsin.

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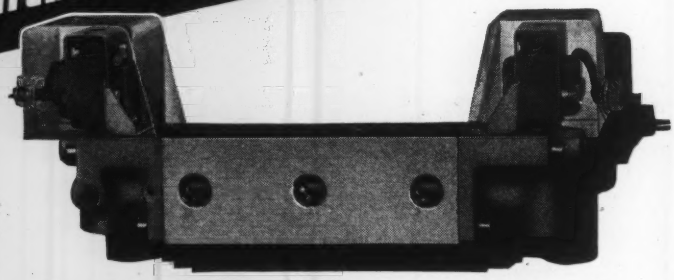
Specify Quick-As-Wink AIR AND HYDRAULIC Control Valves



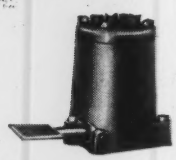
their reduced amperage requirement SIMPLIFIES electrical circuits

● Fast acting and dependable, Quick-As-Wink Control Valves are the proven, time tested answer to efficient — and safe — machine operation. Their short stroke simplifies electrical circuits and avoids destructive impact, minimizing wear and maintenance. They give users millions of cycles of trouble-free, high cycle service. 3/8" to 2" sizes. 2, 3 or 4-way actions. Bucking cylinder or solenoid return. Also available in neutral position types.

There's a Quick-As-Wink Valve for ***every*** machine control requirement. Send for the data sheets. Get full details today!

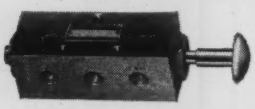
Individual DATA SHEETS for Each Valve — give complete details. Write today!

Hand Operated Air Valves—wide variety of uses. 2-way, 3-way, 4-way neutral position and compound exhaust.

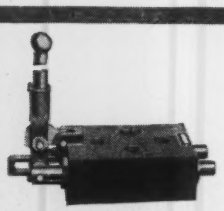


Foot Operated Air Valves—workman has both hands free, speeding production. 2-way, 3-way and 4-way actions

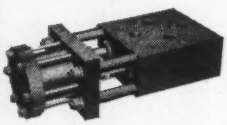
Single Plunger Valves—for air or low pressure hydraulic service. Lever, pilot, cam, diaphragm or solenoid operated. 2-way, 3-way, 4-way actions.



Series "O" and "OE" Valves—for air or hydraulic service up to 125 PSI. Push-pull, cam, pilot, diaphragm and solenoid operated. 1/8" and 1/4" pipe connections. 2-way, 3-way, 4-way and 5-way actions.



Hydraulic Valves—Up to 5000 PSI. Conservatively rated. 1/2", 3/4", 1" and 1 1/2" sizes. 2-way, 3-way, 4-way actions.



Hydraulic Valves—Up to 5000 PSI. Pilot cylinder operated. 1/2", 3/4", 1", 1 1/2" 2", 2 1/2" 3" and 4" sizes 2-way, 3-way, 4-way actions.

Quick-As-Wink Control Valves

THE HIGHER THE PRESSURE THE TIGHTER THE SEAL

Manufactured by **C. B. HUNT & SON, Inc.**

1945 East Pershing Street, Salem, Ohio

teed to 0.0002-in., settings read to 0.0001-in. on vernier scales. Weight, 7000 lb. *Hauser Machine Tool Corp., Manhasset, N. Y.*

PORTABLE SCREW CAPPER: Tightens metallic or molded caps at speeds from 30 to 50 caps per minute. Suspended from ceiling or any overhead point. Weight 4 1/2 lb. Tension adjustment can be preset to any degree of tightness. *Scientific Filter Co., New York, N. Y.*

DIE FILER: Table located at horizontal position by dowel pins in pivot bracket. Table design changed to 4-way, 15-degree tilt. Deep-throated overarm redesigned to permit filing, sawing and lapping with one overarm. Especially suited for needle type files. Ball-jointed type chuck below table permits alignment of files with warped or crooked shanks. Includes two operating speeds and Scotch yoke mechanism to give vertical filing stroke. *Rice Pump & Machinery Co., Milwaukee, Wis.*

SHAPERS: High-speed, all-geared, 18-speed models in 12 and 15-in. sizes. Speeds available, 17 to 200 strokes per minute. Eight feeds provided ranging from 0.006 to 0.053-in. With plain or universal table. Table specifications: horizontal travel, 19 1/2 in.; vertical travel, 12 1/4 in.; max table to ram, 15 1/8 in.; arc of adjustment (universal), 90 degrees. Powered by 2 or 3-hp, 1800-rpm motor. *Western Machine Tool Works, Holland, Mich.*

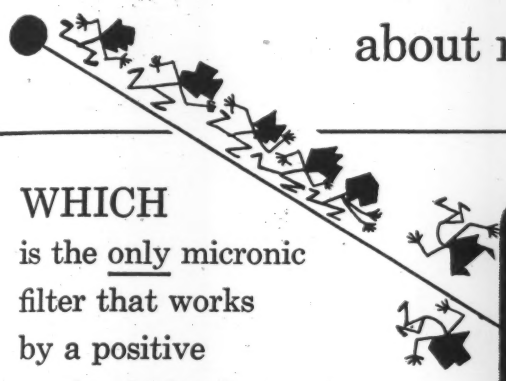
AUTOMATIC NIPPLE THREADING MACHINES: Thread, ream and chamfer nipples in continuous automatic operation, eliminating secondary handling. Mechanism transfers work end-for-end to perform operations on both ends. Standard model produces standard commercial nipples to 6 in. in 1/2 to 3/4 or 1 to 2-in. size ranges. Close-nipple model produces close, short and special short nipples in same two size ranges. Production rates range from 193 to 625 nipples per hour, depending on pipe size. *Landis Machine Co., Waynesboro, Pa.*

TURBINE BLADE GRINDER: Precision two-wheel form grinder for jet engine blades. Uses two 24-in. diamond-dressed grinding wheels to form both sides of blade at once. Machine cycle automatic except for loading and unloading. Parts reciprocate between wheels for predetermined number of strokes at set speed. Wheels feed in after each stroke until finish size is obtained. *Ex-Cell-O Corp., Detroit.*

TUBE BEADING MACHINE: For beading and flaring tubing after it has been bent. Uses spinning principle, with tube held stationary while

Which Design

answers your most important problems about micronic filtration?



WHICH

is the only micronic filter that works by a positive mechanical principle?

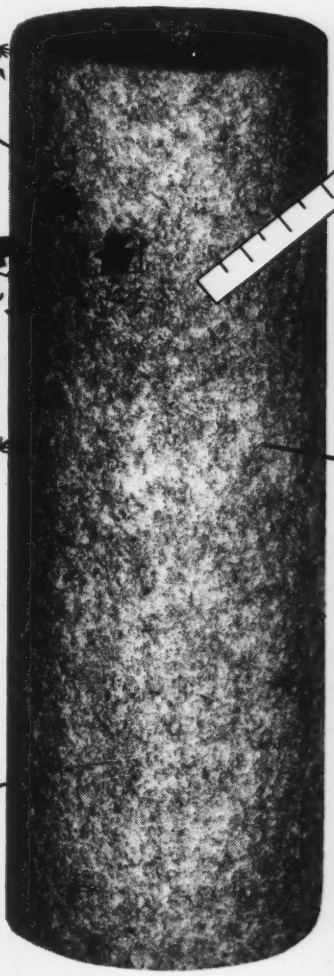
Cuno MICRO-KLEAN cartridge consists of tiny fibres distributed under scientific control and resinous impregnated and polymerized to prevent softening, swelling, hardening, shrinking, rupture or channelling. Solids are simply entrapped in the interstices—no other means is utilized. Densities: 10, 25, 50 microns. Wide range of fluids, flow rates and capacities (from a few to more than 800 gpm). Single or multiple cartridge units.

Your Customers will like . . .

Longer life—Exclusive "graded density in depth" permits smaller particles to penetrate to varying depths—doubles dirt-holding capacity.

Low pressure drop

Changing cartridge is quick—and clean.

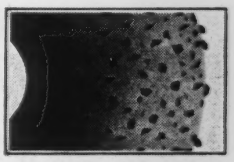


No Fluid Is Better Than Its Filtration

WHICH FILTER

needs the least amount of room?

Cuno MICRO-KLEAN cartridge is utterly simple. It's all filter, no structural components. This means less space needed—and makes *full-flow filtration* practical for either external or built-in applications.



WHICH FILTER

is guaranteed for specific performance?

Felting of fibres is accurately controlled for various densities . . . so that a Cuno MICRO-KLEAN of a given density will positively remove 100% of all solids for which it is rated, plus a large percentage down to 1 micron.

CUNO

Complete Line

Fluid Conditioning

Removes More Sizes of Solids from More Types of Fluids

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WIRE-WOUND Flo-Klean

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- ☐ Fuel Oil
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Company.....

Address.....

City.....State.....

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profile spindle rotates off center to form bead. Profile spindle has ½-in. throw off center. Machine has potential of 1200 cycles per hour. *The Gibbons Machine Co., Tipp City, O.*

VACUUM FLASK SEALER: Joins inner shell to outer shell of vacuum flask at its mouth. Twelve heads located on 45-in. diameter circle. Intermittent motion of turret provided by barrel cam and speed reducer. Head speed adjustable between 30 and 50 rpm. Production, 250 to 400 seals per hour. *Eisler Engineering Co. Inc., Newark, N. J.*

SPECIAL PURPOSE DRILL: For drilling wire holes in hex heads. Employs three Model HH Govro-Nelson automatic drilling units to drill three holes at once. Part is then rotated to drill other three holes. Production rate, drilling six No. 50 wire holes in cap screw with hardness of 25 Rockwell C, 288 pieces per hour. *Govro-Nelson Co., Detroit.*

DRILLING MACHINE: Drills, counterbores and reams steering gear idler and third arm bracket. Two-way, horizontal, 6-spindle machine equipped with 4-position trunnion. Part loaded at first station, drilled from both sides at second, counterbored at third and reamed at fourth station. Interlock prevents counterboring or reaming part not previously drilled. *Lemaire Tool & Manufacturing Co., Dearborn, Mich.*

LATHE: New 12-in. lathe in engine and toolmaker's models. Includes totally enclosed gear box and end gearing, helical gears in headstock, with speed changes made by sliding heavy-sided jaw clutches. Toolmaker's model has quick-clamping type tailstock with tang slot in spindle to facilitate ejection of drill and reamer sockets. *Monarch Machine Tool Co., Sidney, O.*

COPY-MILLER: Universal copy-milling machine for manufacture of small and medium dies weighing to 1100 lb. When copying at 1:1 ratio, pantograph serves only as guide for tracing stylus and cutter head spindle. Parallel guiding device moves longitudinally, transversely and vertically, is counterbalanced by adjustable spring. Cutter spindle speeds stepless between 300 and 9500 rpm. *The Cosa Corp., New York, N. Y.*

POWDER METAL PRESS: Model S-5, 40-ton, cam-operated press for electric motor, radio and television and machine parts, cemented carbides, etc. New design largely eliminates necessity for changing cam inserts. Hand wheel adjustment sets machine for proper compression level of lower punch, automatically adjusting for ejection level. Lower

To an imaginative design engineer...

Spongex®

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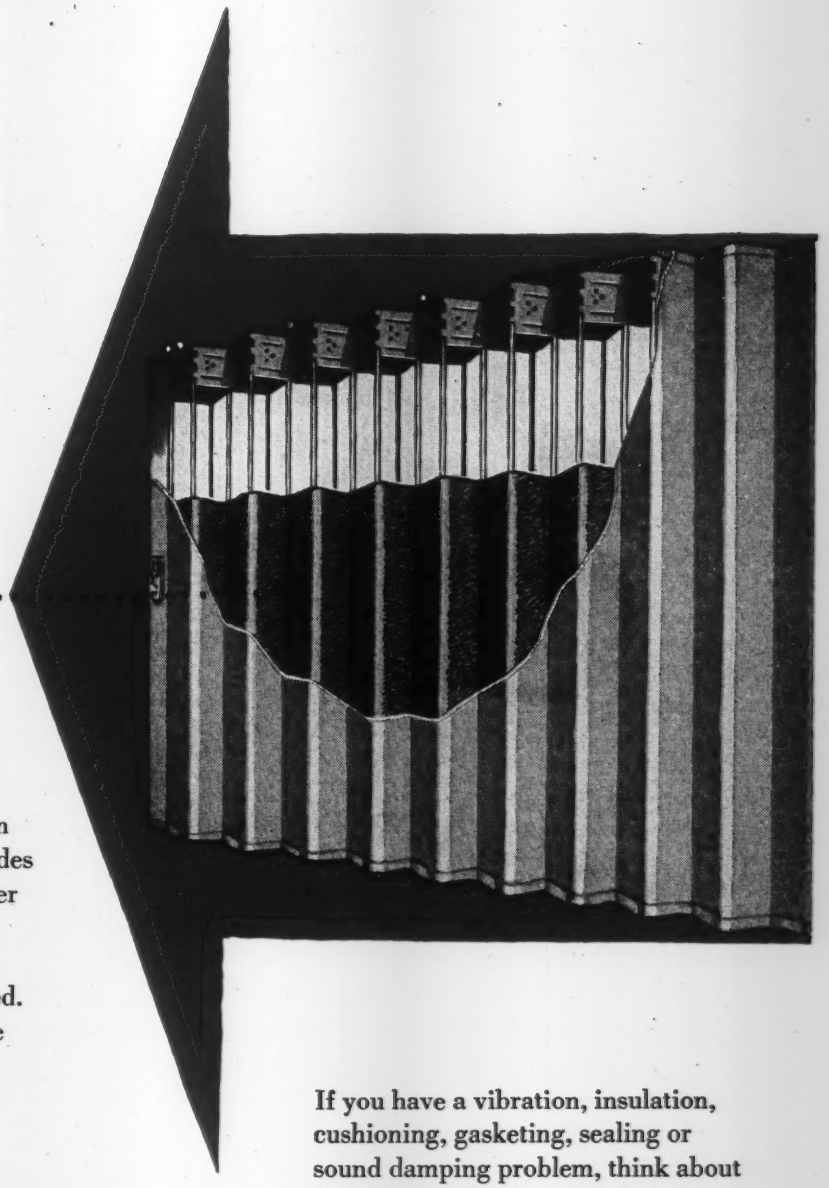
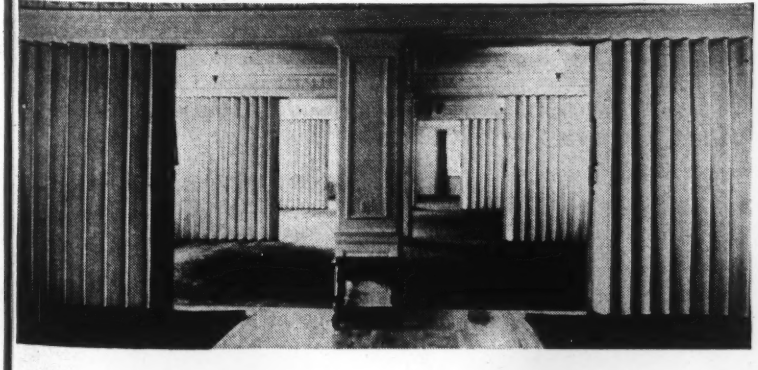
sound insulates Modernfold Doors.....



New Castle Products make the Modernfold accordion-like partition and closure which temporarily divides large rooms into two or more smaller ones. A sound retarding lining for installation in schoolrooms, offices and hotel banquet rooms was needed. In addition to deadening sound, the lining had to be flexible, durable, lightweight, small in bulk, free of dust and dirt, and economical. Spongex cellular rubber, in sheet form, answered their every need.



If you have a vibration, insulation, cushioning, gasketing, sealing or sound damping problem, think about *Spongex*. Cellular rubber does not become a "product" until you make it one in *your* application. We welcome new problems.

Technical Bulletin on Sponge Rubber available upon request.

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★ Gaskets
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★ Conveyor Belts
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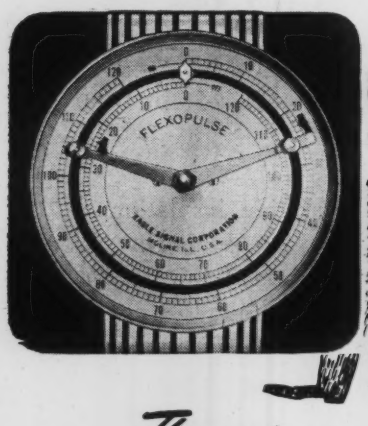

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There is a large, easy-to-read, 6 in. dial which has individual adjustments for setting "ON" and "OFF" time—no gears to change.

a progress pointer to tell you what's happening at all times.

Wide selection of dials:
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plunger spring pulldown eliminates tendency for lower punch to hang in die. *F. J. Stokes Machine Co., Philadelphia, Pa.*

TOOLROOM LATHE: Precision lathe for sensitive instrument work or heavy carbide cutting. Handles 12½-in. swing on 20-in. centers. Headstock back-gearing gives 22 to 400 rpm speeds, belt drive to spindle gives infinitely variable speeds to 3600 rpm. Gear box provides for 72 feeds and 84 threads. *Rivett Lathe & Grinder Inc., Boston, Mass.*

DRAWN SHELL TRIMMER: Trims shells of almost any depth to 14 gage max. Tooling adapter provided for any size or shape shell to 12-in. diameter, making trimming of irregular shell possible to minimum internal radius of 1/16-in. Production capacity, 20 to 500 shells per hour. Machine powered by 3-hp motor. *Dayton Rogers Manufacturing Co., Minneapolis, Minn.*

HAND POWER TOOL: For hand grinding, drilling, polishing, etc. Idling speed, 27,000 rpm; load speed, 20,000 rpm; weight, 2 lb. Chucks interchangeable spring collets of 1/16, 3/32 and ⅛-in. capacity. For work in wood, metal, plastics, etc. *Chicago Wheel & Mfg. Co., Chicago.*

INSTRUMENT LATHE: For high speed, precision tolerance work. Spindle speed, 12,000 rpm; collet capacity, 5/16-in. Features variable speed motor, six-position turret, and 22-in. bed. Attachments available to adapt unit for development or production work. *F. W. Derbyshire Inc., Waltham, Mass.*

SOLDERING GUN: Light-duty gun for radio and television work, for mechanics, electricians, etc. Features dual heat, 100-135 watts, five-second heating time, trigger switch control for adjusting heat to work and two small lights on either side of terminal bars to eliminate shadows and speed up soldering. *Weller Electric Corp., Easton, Pa.*

MANUAL WELDER: Manual Lincolnweld equipment uses lightweight welding gun that feeds electrode and granular flux automatically to give "hidden arc" type weld. Includes control case, wire reel case, conductor cable, welding gun and cable assembly. Operates from standard welding generator. Electrode feed rate automatically controlled by voltage control that maintains constant voltage between electrode and work. *The Lincoln Electric Co., Cleveland, O.*

PORTABLE SPOT WELDER: Self-contained unit converts into foot-operated stationary model by attaching to stand. Weight, 33 lb. Welds to ¼-in. combined thickness of mild or stainless steel, or two pieces of

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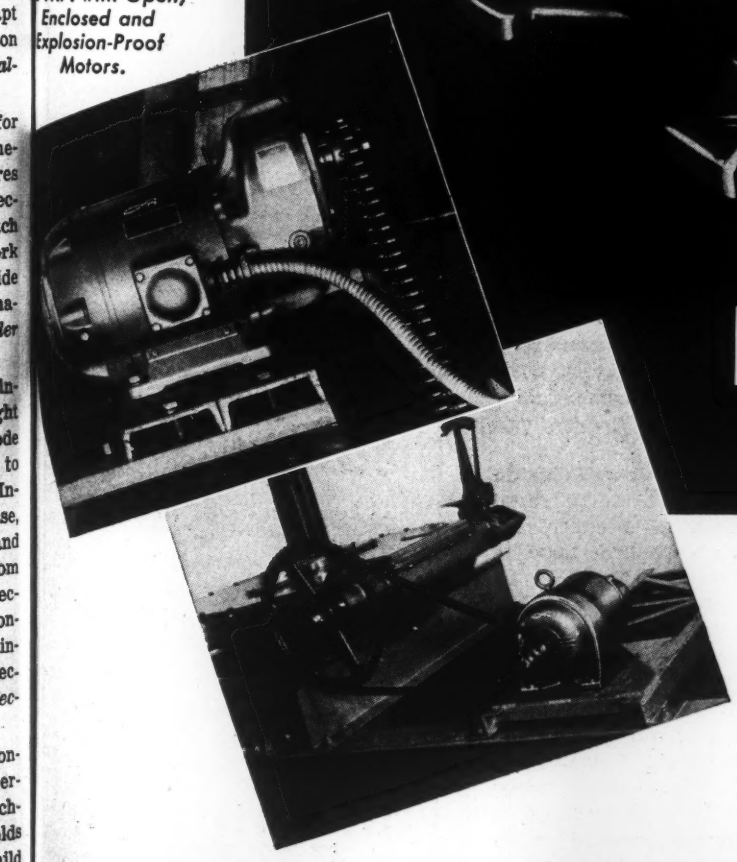

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Typical examples of the many Foote Bros.-Louis Allis Gearmotors now in service. These compact Gearmotors are available in a wide range of sizes from 1 through 75 H. P. with Open, Drip-Proof, Enclosed, Splash-Proof and Explosion-Proof motors.

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16 gage galvanized metal with combined thickness of ⅛-in. Squeezing handle applies pressure to welding tips and turns on current. Copper welding arms available in 6, 12 and 18-in. lengths. *Greyhound A. C. Arc Welder Corp., Brooklyn, N. Y.*

Materials Handling

POWER CONVEYOR: Adjustable, portable unit for handling small stampings, castings, borings, turnings, chips, etc., where working heights and areas change frequently. Height adjustable through 30-in. range by double-screw center lift. Pitch adjusts from horizontal to 45 degrees by crank. In 4, 6, 8, 10, and 12-ft lengths with 12, 18 or 24-in. belt widths. Includes charging chute. Motor obtainable to suit requirements. *May-Fran Engineering Inc., Cleveland, O.*

HAND CHAIN HOISTS: Capacities, ½, 1, 1½ and 2 tons. Smallest model weighs 37 lb, has 95 per cent efficiency, permits raising 1000 lb three ft in 20 seconds. Springs force 6-tooth pawl into engagement with 24-tooth ratchet for positive automatic braking. Aluminum castings used to reduce weight. *Philadelphia Div., The Yale & Towne Manufacturing Co., Philadelphia.*

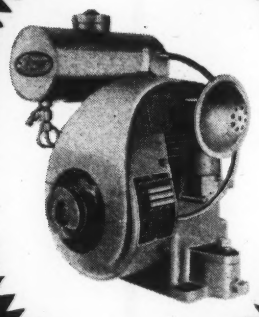
LIFT TRUCKS: In sit-down and stand-up models. Capacity, 2000 lb on 15-in. load center. Powered by 3-cylinder gas engine with 3-in. bore, 3½-in. stroke. Sit-down model, with 83-in. mast, has 108-in. lift height, 134-in. mast-extended height, 57-in. free lift. Length, less forks, 65 in.; width, 33⅛ in.; turning radius (outside) 63 in. Weight of both models, 3300 lb. *Mobilift Corp., Portland, Oreg.*

AUTOMATIC PRESS UNLOADERS: "Iron Hand" for unloading stamping presses. Strokes per minute range, 12 to 20. Power cylinder strokes of 10, 16, 20 and 24 in. Limit switch stops press midway in downstroke if jaws do not retract to rest position due to failure in air line. *Sahlin Engineering Co., Birmingham, Mich.*

HOIST: Hydraulically powered, ¾-ton shop or truck hoist. Lifts rated load 77½ in. in 45 seconds. By using short hookup, lifts 2000 lb 52 in. in 45 seconds. Powered by double-action hand-operated hydraulic pump. *Unit Manufacturing Co., Minneapolis, Minn.*

HAULAGE VEHICLES: In tip or dump models, each with bulk load capacity of 40 cu ft and weight carrying capacity of 4000 lb. Powered by gas engine. New two-speed transmission gives 14 mph in high, 8 mph in low, either forward or reverse. Shortened wheelbase has reduced overall length and decreased

choose



THE *LAUSON Long Life* ENGINE

ILLUSTRATED:
• Lauson's new bantam beauty — Model LMH. Weighs only 22 lbs.! 1 HP. Other sizes to 6 HP. All Lauson engines are 4-cycle, air-cooled!

• Insist on Lauson — the engine that's *first in quality!*
Here are just a few of the superior construction features which make Lauson the *Long-Life* engine: Connecting rods have replaceable crank-pin bearing liners.

• Precision ball bearings on both ends of crankshaft.

• Fly-ball governor running in oil.

• Automotive-type float feed carburetor.

• Lauson original design provides direct stream of cool air over both valves simultaneously.

• Always choose Lauson engines — a better buy because they're better built!

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A Division of Hart-Carter Company
In Canada: Hart-Emerson Ltd., Winnipeg, Canada

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AN IDEAL LAMINATED MATERIAL for CONTACTORS

The many advantages of this new material for electrical contacts are being utilized by General Electric in its latest line of a-c motor starters.

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Costly assembly operations are reduced to blanking costs . . . no waste of precious metal.

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Wearing surfaces on base metal are rolled to required temper.

IMPROVED CONTACT TIPS

Precious metal is hard rolled to produce dense, hard contact surface with highly desirable electrical and mechanical qualities.

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"Raised-Lay" consists of one or more strips of silver or other precious metal bonded to the base metal in the form of a ridge or bar, providing very definite advantages for both manufacturers and users of contactors.

"Raised-Lay" is available in any strip width and thickness in both single, double and double-double combinations and can also be made in any desired combination of precious and base metals to specifications. We are also prepared to furnish similar contact material in the form of "In-Lay", "Edge-Lay" and "Over-Lay".

In addition, we offer a complete blanking and fabricating service on "Raised-Lay" and our other contact materials to meet any requirements.

Your inquiries are cordially invited and will receive our prompt and interested attention.

*Patents applied for covering both the material and the method of application.

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turning radius under previous model. *Clark Equipment Co., Industrial Trucks Div., Battle Creek, Mich.*

HORIZONTAL CONVEYOR: Handles distributed loads to 4000 lb. Roller sections have upper level of carrier rollers, lower level of adjustable pressure rollers, and 8-in. canvas belt running between two levels serving as friction drive for surface rollers. Available in lengths to 150 ft, four standard widths from 15 to 24 in. *The Rapids-Standard Co. Inc., Grand Rapids, Mich.*

ADJUSTABLE FORKS: For lift trucks. Operator changes fork spacing hydraulically without leaving seat. Can be changed with truck in motion. For operations requiring handling of pallets or materials of different widths. *Towmotor Corp., Cleveland, O.*

FORK LIFT TRUCKS: Gas-powered Carloader modified to include Dynatork drive. Engine power transmitted to drive wheels through air gap by magnetic induction. Conventional transmission replaced by constant-mesh, two-speed forward and reverse gearing. Permits reversal of direction without bringing machine to full stop. *Clark Equipment Co., Industrial Truck Div., Battle Creek, Mich.*

TUMBLE HOPPERS: For feeding and orientation of most cylindrical work pieces. Rotary type cast aluminum hoppers available in 8, 12, 16 and 24-in. diameter sizes. Motor-driven units include gear reducer arranged to give variable speed to rotor. *Kraus Design Inc., Rochester, N. Y.*

HIGH-LIFT TRUCK: Hand-operated unit for work positioning, stacking, tiering, handling dies, etc. Capacity, 1000 lb. Standard elevated heights, 48 and 60 in. Telescopic frame available as accessory increases elevated height to 108 in. Hydraulic hand pump standard, can be equipped with electric powered pump for lifting operation. *Lyon-Raymond Corp., Greene, N. Y.*

Plant Equipment

DUST COLLECTOR: For separating industrial dusts from air stream and discharging toxic fumes outside. Available as standard, pretested units having capacity of 10,050 cfm. Powered by 15-hp motor driving paddle wheel fan, unit has static suction lift of 4 in. on 15-in. inlet. Size of outside discharge, 18 in. *Aget-Detroit Co., Ann Arbor, Mich.*

UTILITY STEAM BOILERS: For plastic presses, paint heaters, wallpaper steaming, sterilizing, laboratory work, etc. Generate steam at 98½ per cent efficiency. Dimensions, 14 by 22 by 28 in. On 110-v a-c, units

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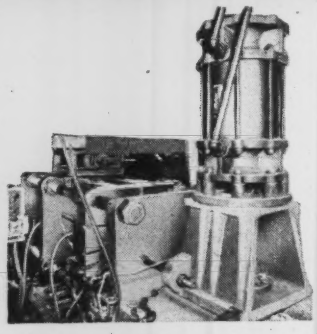

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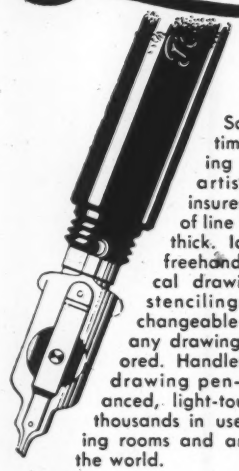

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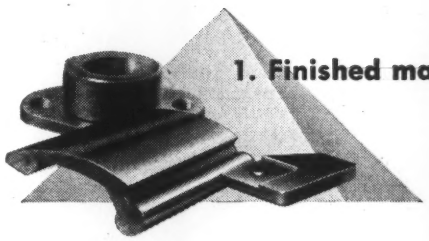
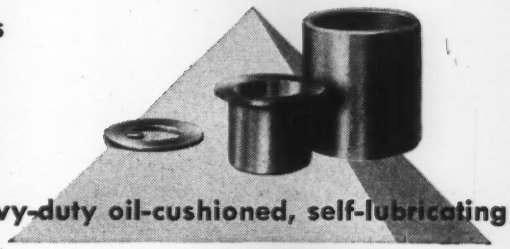
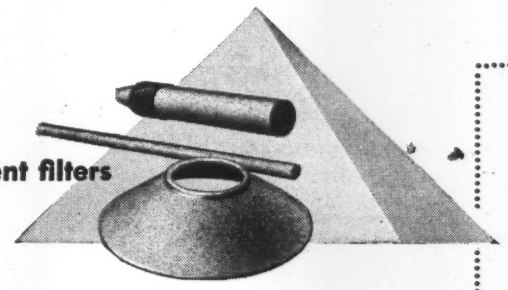


1. Finished machine parts

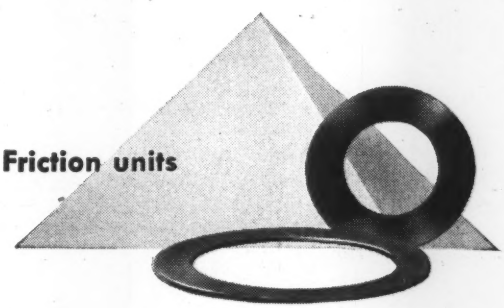


2. Heavy-duty oil-cushioned, self-lubricating bearings



3. Permanent filters

4. Heavy-duty oil-cushioned, self-lubricating cored and bar stock



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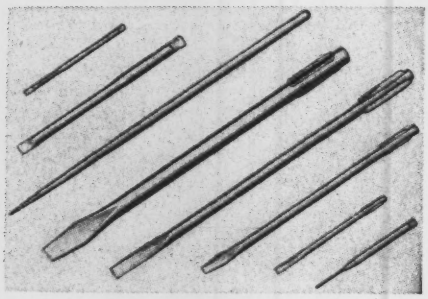

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Plant Equipment

DOOR OPENERS: For raising and lowering commercial and industrial doors or gates. Friction drive gives maximum protection to personnel and property. Pushbutton operated by remote control. Models for balanced doors or sliding gates. Sizes to handle door areas of 160, 200 and 250 sq ft, max door weights of 480, 600 and 750 lb. *Robot Appliances Inc., Dearborn, Mich.*

IMPACT TOOLS: Air-operated units in sizes for nut running to ⅜ and ¾-in. bolt size. Portable tools have pistol grip, grooved reverse caps for easy reversal. Large unit weighs 11¼ lb, measures 10⅞ in. long. Smaller unit weighs 5¾ lb, measures 8$^{9}/_{16}$ in. long, may be used for drilling to ⅜-in. diameter, step drilling to ¾-in., reaming to $^{9}/_{16}$-in. diameter, tapping to ⅝-in. diameter, stud removing, etc. *Ingersoll-Rand Co., New York, N. Y.*

NUT SETTERS: Two angle type units provide high torque for medium-sized nuts. Designed with 12:1 gearing in gear case which, with 1:1 and 2:1 gearing in angle attachments, provides torques of 30 and 60 ft-lb. Angle attachments adapt tools to work in close quarters. *Keller Tool Co., Grand Haven, Mich.*

COMPRESSOR: Capacity, 36 cfm at 80 psi. Light, compact, gas-driven unit weighs 265 lb ready to run, stands 32 in. high on 27-in. diameter baseplate. Wheelbarrow mounting provides portability. Horizontal arrangement of three power cylinders and three air cylinders around vertical single-throw crankshaft eliminates need for flywheel. *Ingersoll-Rand Co., New York, N. Y.*

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